



Impact of microvibrations on low temperature instruments dedicated to the study of the universe

Thomas Adam

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Pour obtenir le grade de

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Impact des microvibrations sur les instruments très basses températures dédiés à l'étude de l'univers

Impact of microvibrations on low temperature instruments dedicated to the study of the universe

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If you have sharp sight, use it: but, as the poet says, add wise judgement.

– Marcus Aurelius

To my nan, et à mon papi.

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I am now convinced, as with many of life's endeavours, that with a bit of *tête dans le guidon* there is often indeed a fantastic view above one's own intellectual *mer de nuages*!

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Introduction

As satellite performance requirements for disruptive science missions have continuously increased over the years, the impact of microvibrations on system design could become ever more critical. The performance of specific low-temperature instruments for such missions are very sensitive to the thermal stability of the coldest cooling stage, leading to stability requirements as low as 1 μ K. This allows them to detect incoming light (say from X-ray, infrared or microwave radiation) base on thermal power (or thermal energy) measurement, with resolutions as low as several attowatts.

A source of parasitic heat is the dissipation due to microvibrations, directly threatening the performance of said science missions. Besides the many existing microvibration sources aboard a satellite (reaction wheels, mechanisms and thrusters to name a few), a trend towards more mechanical cryocoolers for low temperature missions (gradually replacing cryogen baths) is also increasing microvibration disturbance.

The objectives of this PhD were to characterise the impact of dissipation due to microvibrations, through the study of elementary cryogenic devices as well as more complex cryogenic systems. The test objects chosen were a sub-Kelvin cooler coupled to a focal plane assembly (FPA) mockup and one of its subcomponents – a device ubiquitous in sub-Kelvin space missions – that we studied separately. The sub-Kelvin cooler is based on an adiabatic demagnetization refrigerator precooled by a sorption cooler. It cools the FPA mockup from 2 K to 50 mK. Together they form the complex sub-system briefly studied.

Within the first months of the thesis, the complex sub-system was available for us to carry out tests. Needing no additional modifications to study microvibration dissipation, it was an ideal opportunity to test a relatively mature full-scale cryogenic sub-system that was both thermally and mechanically complex. It is shown mounted onto a helium bath cryostat in Figure 0.1.

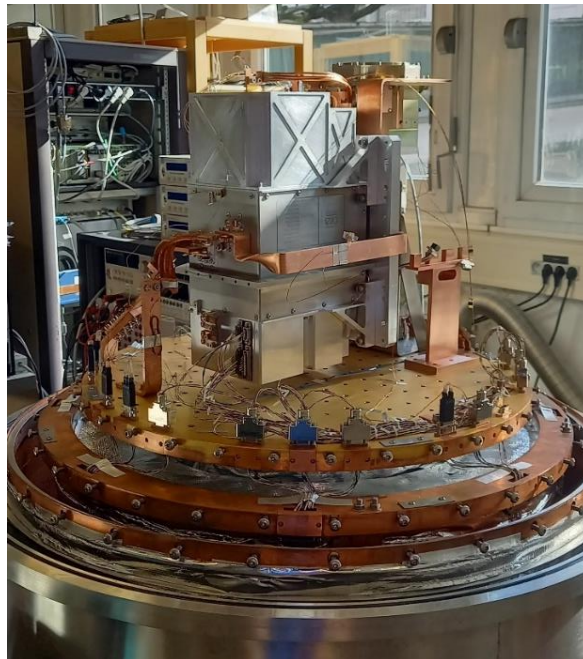


Figure 0.1 – Sub-Kelvin cooler (front) coupled to FPA mockup (top back)

However, though worthwhile observations were made, they also highlighted the difficulty of studying microvibration dissipation in complex systems. We therefore chose not to present the observations made with the sub-Kelvin cooler and to focus on a simpler object, a gas-gap heat switch, to gain a more fundamental understanding of the phenomena involved in microvibration dissipation. It is shown in Figure 0.2. We shall see that that this device too, is more complex than we had anticipated.



Figure 0.2 - Gas-gap heat switch (main test object)

My PhD was carried out in the cryocoolers and space cryogenics team of the DSBT (*Département des Systèmes Basses Températures*) of the CEA (*Commissariat à l'Energie Atomique*) in Grenoble, France. It was co-financed by the CEA and CNES (*Centre National d'Etudes Spatiales*).

My work is largely focused on experiments and gathering valuable data. The complexity of reproducible experiments and of singling out the main driving parameters have been at the centre of my work

The first chapter presents the thesis topic, beginning with the scientific context, from microvibrations to cryogenics. Some literature in the field of microvibration dissipation in space will then be presented. The latter half of the chapter presents the damped mass-spring model used to estimate microvibration dissipation and the thermal model. These are used to support the experimental observations presented in later chapters. We close the chapter outlining the main objectives of the thesis.

The second chapter presents the working principle and characterisation of our main test object (a gas-gap heat switch), and the cryogenic environment used for this thesis. We also present the wide range of mechanical and thermal equipment used, from sensing to actuation. The purpose of this chapter is to raise the reader's awareness of the complexity of inducing sub-mg microvibrations in a controlled manner, and measuring sub-microwatt microvibration dissipation in a small device at cryogenic temperatures.

The third chapter presents a characterisation of our measurement environment and our main methods for measuring microvibration dissipation. In the final part of the chapter we discuss interactions between our test object and cryogenic test environment that affected our measurements.

The fourth chapter presents the results obtained with our main test object. We detail the different test configurations and analyse observations of our various test campaigns. The purpose of this chapter is to explore the nature and location of microvibration dissipation in and around our test object.

To close this dissertation, we discuss the main conclusions of this thesis and the perspectives of continued research in this promising field, both within the CEA/DSBT lab and in the wider space community.

General presentation

Early observations of motion in nature, from the swaying of trees to the swell of the oceans and the motion of celestial bodies, led humankind to question the behaviour of its surroundings. The study of motion, as ubiquitous at the beginning of the universe as today, marked the birth of classical mechanics. The study of the macroscopic motion of objects is a core branch of physics.

The study of classical mechanics dates back as far as the 4th century BCE, beginning with foundational contributions from Aristotle, formulating the concepts of motion and force. Since then, the field has expanded thanks to renowned physicists and polymaths. To name a few, Galileo formulated the principle of inertia, Kepler famously his 3 laws of planetary motion and Newton his own laws of motion as well as – legend has it with the help of an infamous apple – the law of universal gravitation.

Mechanical vibration, itself a branch of classical mechanics, specifically describes the oscillatory motion of objects about a point of equilibrium. Notable contributions to this particular branch include the nature of pendulums by Huygens, the nature of vibrating strings and membranes by Euler, the formulation of the wave equation by d'Alembert and the work on wave propagation in fluid dynamics by Bernoulli.

In some instances, vibration is desirable (e.g. in string instruments, loudspeakers, vocal cords, eardrums, etc.) but in most cases, it is not. Vibration typically accelerates the wear-and-tear of a system and reduces its energy efficiency. For example, for vehicles traveling on surfaces, through air or through space, vibration will cause material fatigue over time, leading to breakage and system failure. It is as crucial to address the *generation* of vibrations (e.g. from motors or actuators) as the *susceptibility* to vibrations (e.g. of measurement equipment). Despite actively mitigating both, neither is fully eliminated.

The residual generated vibrations are known as microvibrations. The term microvibration refers to small-scale mechanical vibrations due to dynamic interactions. Their amplitude range is typically micro-g to milli-g and frequency range 1 Hz to 1 kHz.¹ Understanding their impact is critical in systems housing precision instruments and sensitive equipment, where even minor disturbances can affect performance.

Ground application examples include optical systems (sensitive to pointing stability) and MEMS systems (sensitive to small dynamic disturbances), though microvibrations are of particular interest in space applications. Systems operating in microgravity environments and in the vacuum of space often show little damping, potentially increasing system susceptibility to microvibrations.

Though the applications of microvibration extend beyond space, the motivations for this thesis stem from recent microvibration challenges, and the future ambitions, of low-temperature space missions. To introduce the scientific context of this thesis, in this chapter we will delve into the field of microvibrations in space and later its intersection with the field of cryogenics in space.

¹ (Liu, Jing, Daley, & Li, 2015)

1.1 Context

1.1.1 Microvibrations in space

Microvibration sources in space include mechanisms such as Reaction Wheels (RW), Solar Array Drive Mechanisms (SADM) or Antenna Pointing Mechanisms (APM) to name a few.² Additional disturbance sources include thrusters, instrument-specific mechanisms such as scanning mirrors or filter wheels and mechanical cryocoolers for cryogenic instruments. Rarer sources include thermomechanical effects from solar shades and solar arrays subjected to changes in thermal loading from passing in and out of sunlight.³

The history of microvibrations in space, related to the public domain, dates as far back as the 1970's when NASA first conceptualised the Large Space Telescope that was later to become the famous Hubble Space Telescope (HST). State-of-the-art space observatories such as the HST or more recently the James Webb Space Telescope (JWST) host extremely sensitive payloads that impose stringent requirements on the permitted pointing instability – known as line-of-sight (LOS) jitter – caused by microvibrations.³⁻⁴



Figure 1.1 – Hubble Space Telescope

The highest requirement for the HST is less than 7 milli-arcseconds over a 24-hour period⁴, an angular resolution equal to seeing a beetle scrambling the cliffs of Dover in the UK, from Grenoble in France. For the JWST it is almost half, at less than 3.7 milli-arcseconds of LOS jitter. It is easy to grasp the sensitivity of such systems to microvibration and why the topic has gained increasing attention over the years.³

This trend is not least driven by national space agencies such as the National Aeronautics and Space Administration (NASA) and the European Space Agency (ESA) pushing for higher performing payloads and instruments, particularly for scientific and Earth observation satellites. Microvibration has thus grown to become a system-level spacecraft engineering problem, involving multiple disciplines: structures,

² (Smet & Patti, 2018)

³ (Dennehy & Alvarez-Salazar, 2018)

⁴ (Jedrich, et al., 2002)

mechanisms, AOCS systems and more recently cryogenics. The latter is a branch of thermodynamics, the study of the relations between heat and other forms of energy (e.g. mechanical). As we will see next, it is the intersection of cryogenics in space and microvibrations where the motivation of this thesis lies.

1.1.2 Cryogenics in space and microvibrations

We shall now see the heritage and long term trends of cryogenics in space, further highlighting the importance of understanding the impacts of microvibrations. The intersection between space cryogenics and space microvibrations is also discussed here. In section 1.1.3, we then present the core topic of this thesis, namely the study of the thermal impacts of microvibrations at low temperatures.

1.1.2.1 Cryogenics in space

Cryogenics in the field of space predates the relatively nascent field of microvibrations. Cryogenics made its first entry into space at the launch pad, with the development of cryogenic storage of liquid oxygen and liquid hydrogen, now common rocket propellants.

The space race of the 20th century relied heavily on cryogenic fuels.⁵ They propelled the first manned space missions, launching the first human into space in 1961, Yuri Gagarin. NASA relied on cryogenic fuels in the Apollo program for its Saturn V rocket, landing humans on the moon in 1969, as well as for its Space Shuttle era. More recently, Europe's flagship Ariane 6 launcher – whose maiden flight was in 2024 – still uses cryogenic fuels. Still today, without cryogenic technology there is no access to space.

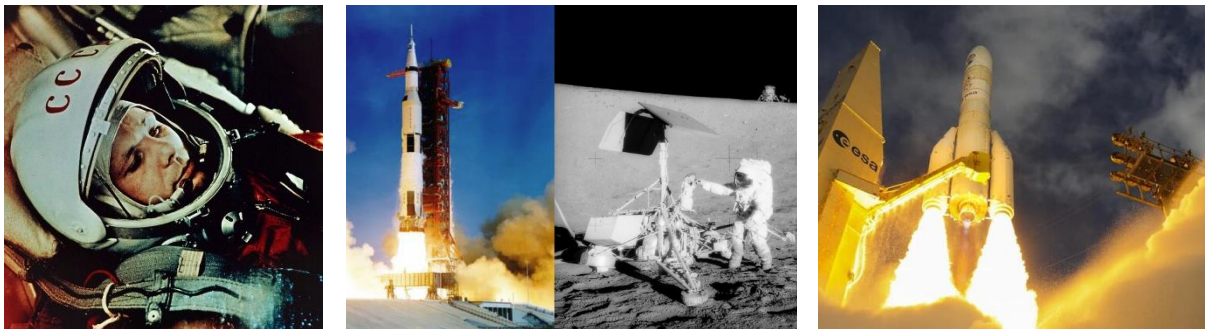


Figure 1.2 - First human in space 1961 (left), first human on the moon 1969 (middle), Ariane 6 maiden flight 2024 (right)

Cryogenics in space has expanded beyond cryogenic-liquid rocket propellants, particularly for military, Earth observation and science missions. Aboard the International Space Station (ISS), say for storing biological samples, but particularly further afield has cryogenics become essential. By far the largest application of space cryogenics is for infrared Earth observation satellites and missions addressing the science of the universe. The latter cover several spectral regions, such as the infrared, microwave, X-ray and gamma ray wavelengths. Examples include the Meteosat Third Generation (MTG), Planck, Herschel, JWST, Astro-H and Integral missions. The cryogenic technologies of some of these will now be discussed.

⁵ (Ross, 2007)

1.1.2.2 Earth observation missions

An example of cryogenics for Earth observation satellites is the new generation of Europe's flagship meteorological satellites, Meteosat Third Generation (MTG). They form a constellation of 6 satellites in geostationary equatorial orbit. Stored cryogen-based cooling would not allow each satellite to achieve its nominal lifespan of 8.5 years, nor the 20 years of planned operation for the full MTG program. The Large Pulse Tube Coolers (LPTC) aboard each satellite (in part developed by CEA/DSBT) cool their detectors to 50 K, allowing the MTG system to study Earth's climate in the infrared region.⁶ The LPTCs are equipped with active vibration control, mitigating exported microvibration disturbance from its compressor. This limits but does not eliminate LOS jitter, a critical performance criterion for Earth observation satellites that generally require high pointing accuracy. The susceptibility of the MTG system to microvibrations was further studied through the microvibration characterisation of some of its mechanisms.⁷

1.1.2.3 Space science missions

Facing away from Earth and towards outer space, two unique European-led missions relying on space cryogenics are worth mentioning: the Herschel and Planck space observatories. They were launched together in 2009 by ESA, studying the far infrared wavelengths and microwave background respectively.

The Herschel mission carried the largest telescope ever flown in space, with a primary mirror diameter of 3.5 meters, until the launch of JWST in 2021. Cooling of the detectors to 300 mK, of 2 of its 3 instruments, relied on 2300 litres of superfluid liquid helium and an adsorption cooler stage developed and manufactured by CEA/DSBT, allowing them to study in the far-infrared region and make highly sensitive observations of some of the coldest and most distant objects in the universe.⁸ After exhausting the liquid helium needed for cooling its detectors, the mission ended after almost 4 years of operation in 2013.

The Planck mission observed in the microwave region. Its detectors were cooled through successive cooling stages. Passive cooling using thermal shields cooled it down to 50 K, followed by 3 different types of active coolers: a hydrogen sorption cooler down to around 20 K, a Joule-Thomson (JT) cooler to 4.7 K and an open dilution refrigerator, to cool its most sensitive detector to 100 mK. These low temperatures allowed it to map the Cosmic Microwave Background (CMB) with unprecedented accuracy.⁹ At such temperatures, it was the coldest known object in space, until the exhaustion of the helium gas reserves used to drive the dilution refrigerator, marking its end of life after over 4 years in operation also in 2013.

Space telescopes studying the X-ray region are also a prime example of the need for space cryogenics. Many celestial objects, such as stars and black holes emit X-ray radiation when undergoing violent processes. Hot gases in galaxy clusters, the largest structures in the universe, emit X-rays too.

Certain spectral regions are absorbed by Earth's atmosphere however, as shown by Figure 1.3. X-rays (0.01 nm to 10 nm) are fully absorbed, as well as some infrared (780 nm to 1 mm) and some microwave (1 mm to 1 m) regions. On the contrary, we note that some radio waves (> 1 m) transmit very well through the atmosphere, the reason for which they have historically been used for long distance communication.

⁶ (Wiertz & Urbano, 2016)

⁷ (Onillon, Adam, & Borque Gallego, 2021)

⁸ (Jewell, et al., 2010)

⁹ (Planck collaboration, 2011)

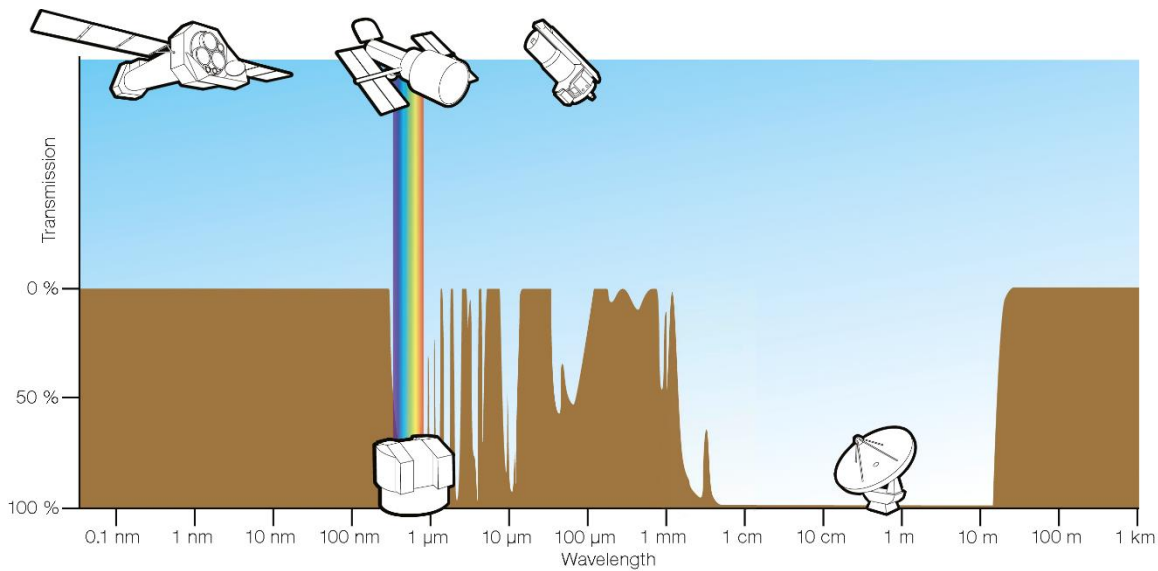


Figure 1.3 – Transmission through the atmosphere as a function of wavelength¹⁰

Given this opacity, X-ray (and some infrared and microwave) detectors must be placed in space to observe the universe. Three X-ray space telescopes are worth mentioning as they use very low temperature detectors and could show susceptibility to microvibrations: Astro-H, XRISM and Athena.

The Astro-H mission, later renamed *Hitomi*, was launched in 2016 by the Japanese Aerospace Exploration Agency (JAXA) and was the 6th of JAXA's X-ray astronomy satellites. The most sensitive detector aboard was cooled to 50 mK by a pair of adiabatic demagnetization refrigerators (ADR). This was colder still than its ancestor, Astro-E2 (launched in 2005), that was cooled to 60 mK and was the first low temperature X-ray detector in orbit. Cooling Astro-H from 300 K down to the ADR's heat sink was achieved by a combination of 30 L of superfluid liquid helium and multiple cryocoolers. Cryogenic redundancy allowed it to operate cryogen-free if needed.¹¹ Unfortunately contact was lost with the satellite early on in the mission due to an uncontrolled spin rate leading to satellite break up.

The XRISM mission led by JAXA is considered the successor to Astro-H. It was launched during my thesis in 2023 and is part of the next generation of X-ray telescopes after Chandra (NASA) and XMM-Newton (ESA), both launched in 1999 and still in operation. Without it, astronomers would have faced a blank period of X-ray observation until the launch of Athena. It too used a microcalorimeter cooled to 50 mK. Its first observations of galaxy nuclei, dark matter and the formation of the universe are promising.

The Athena mission, standing for Advanced Telescope for High-Energy Astrophysics, is a future X-ray observatory planned for launch in 2037 by ESA. Athena will house 2 instruments, one of which being a cryogenic X-ray imaging spectrometer: the X-ray Integral Field Unit (X-IFU). In 2023, ESA request for significant cost reductions required the thermal design of Athena to be adapted. A by-product of this was a welcome reduction of microvibration disturbance sources thanks to fewer cryocoolers. The telescope is now to be cooled passively to 50 K, then to 5 K using mechanical coolers. The final cooling stages, from 5 K to 50 mK at the heart of the X-IFU instrument, will be provided by a series of 5 ADRs. This key cryogenic technology will ultimately allow X-IFU to observe diffuse sources such as hot gases in galaxy

¹⁰ (Granato, 2010)

¹¹ (Fujimoto, et al., 2010)

clusters. Together, the two instruments aboard Athena will help answer questions such as how matter assembles into large-scale structures and how black holes grow and shape the universe.¹²

1.1.2.4 General trends

From Earth observation satellites to infrared, microwave and X-ray space telescopes, these examples highlight that many detectors attain their high sensitivities and low background noise levels using very low temperatures. With a push towards longer lifespan and lower temperatures for cryogenic detectors, stored cryogens and passive cryoradiators are being complemented (or even gradually replaced) by mechanical cryocoolers. These trends in space cryogenics directly impact both the *generation* of microvibration disturbance (more cryocoolers) and the *susceptibility* to them (lower detector temperatures).

We have thus far discussed the generation of microvibration and the mechanical susceptibility to them. This prepares the groundwork for discussing the core topic of this thesis: the *thermal* susceptibility to microvibrations at low temperatures.

1.1.3 Cryogenic detectors and microvibrations

We have seen in section 1.1.1 that several sources of microvibration disturbance may exist and that the classical microvibration problem (LOS jitter) reflects a purely *mechanical* susceptibility to microvibration.

It was further discussed in section 1.1.2 that a trend in space cryogenics towards more cryocoolers may also add to these microvibration disturbance sources. On the Hubble Space Telescope in fact, existing cryocooler technologies were even excluded due to the mission's stringent pointing stability requirement. This led to a redesign (using gas bearings) of the cryocooler used to cool its NICMOS instrument.⁴

In this section we shall see the growing cause for concern of the *thermal* susceptibility to microvibrations, becoming critical at cryogenic temperatures. Specifically, we wish to study the thermal susceptibility of cryogenic systems to microvibrations. Our work is relevant to sub-Kelvin thermal detectors in particular, which are used in several of the astrophysics missions discussed.

The basic functioning of such detectors will first be presented, before discussing why their performance may be affected by the thermal disturbances caused by microvibrations.

1.1.3.1 Working principle of cryogenic detectors

Many of the astrophysics missions discussed use low temperature thermal detectors. By design they show high thermal susceptibility. They consist of arrays of thermally insulated membranes, which heat microscopically when energy is deposited on them. Incident (e.g. infrared, microwave or X-ray) radiation is observed by measuring their temperature fluctuations, by means of a thermometer.

Calorimeters and bolometers are both thermal detectors, though a calorimeter measures the *energy* of single particles absorbed, whilst a bolometer measures the *energy flux* (power) of incident radiation.¹³

¹² (Prouvé, et al., 2017)

¹³ (Kraus, 1996)

Figure 1.4 more closely depicts the functioning of a calorimeter, though bolometers could also be impacted by the dissipation due to microvibrations.

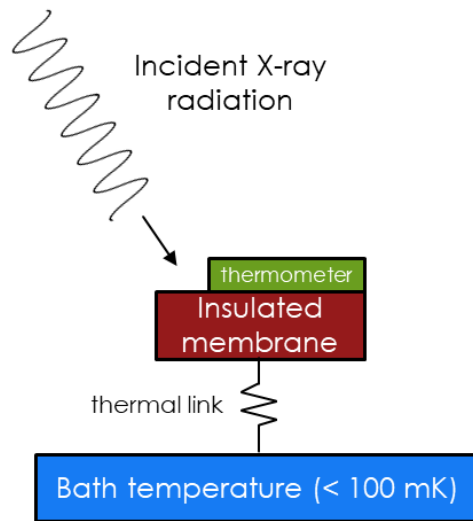


Figure 1.4 – Working principle of a calorimeter detector used in low temperature X-ray instruments

The detector is thermally coupled to the coldest cooling stage and generally cooled to below 100 mK, such as for some of the X-ray telescopes previously presented. The lower the temperature of this stage, the greater the sensitivity of the detector to energy fluctuations due to incident radiation, especially by way of lower thermal noise, but also lower thermal conductivities and heat capacities.

The Herschel, Planck and Astro-H missions housed low-temperature thermal detectors operating at sub-Kelvin temperatures. The Athena mission is to house one too. Indeed sub-Kelvin detectors operating in the infrared or X-ray wavelengths are a key feature of space missions addressing the science of the universe. These detectors must generally be cooled to around 50 mK to achieve high performance.⁵ Astro-H used a so-called doped silicon microcalorimeter, whilst Athena will use a TES microcalorimeter.¹⁴

In the case of detectors based on transition-edge sensors (TES), cryogenic temperatures also enable high measurement sensitivity by exploiting the superconducting phase transition. TES thermal sensors display an electrical resistance that is highly sensitive to their temperature at their superconducting phase transition. Placing a superconductor at its transition temperature, the temperature between the normal and superconducting phases, makes TES-based thermal detector highly sensitive to the smallest temperature (i.e. heat load) variations. The superconducting phase transition occurs at very low temperatures, hence the need for cryogenic technology.

The thermal *stability* of low-temperature thermal detectors is also crucial to their performance. This is because fluctuations in the bath temperature will translate to fluctuations in the temperature of the insulated membrane (i.e. induce thermal noise). These fluctuations could be misinterpreted as a measure of incident radiation by the thermometer. Temperature instability of the bath temperature may therefore degrade the energy resolution of incident photons (or radiation, in the case of bolometers).

We have now seen the working principle of low-temperature thermal detectors, their need for cryogenic temperatures and their by-design sensitivity to parasitic heat load fluctuations, affecting thermal stability.

¹⁴ (Charles, et al., 2016)

The latter is particularly true for low-temperature thermal detectors such as those aboard specific past and future astrophysics missions discussed. We shall now see through examples how through this thermal susceptibility, said sub-Kelvin detectors may be impacted by the dissipation of microvibrations.

1.1.3.2 Impact of microvibrations on detector stage

As discussed, by their nature thermal detectors are sensitive not only to incident radiation but also to other stray energy sources, for example the dissipation of microvibrations. These could affect detector performance by dissipating at the cold stage but also at the detector level. This thesis however, has focused on the study of the impact of the dissipation of microvibrations at the cold stage, rather than inside the detector itself. It is also important to highlight at this stage the separate impacts of *constant* heat load versus *fluctuations* of heat load, due to microvibration dissipation at cold stage level.

A constant heat load reduces available cooling power but leaves detector performance unhindered. Only if it exceeds available cooling power (i.e. the detector stage heats up) is detector performance affected. During in-orbit operation on the Planck satellite, when the 4 K JT cooler unexpectedly shut down, a drop in heat load of 26 nW was observed at the 100 mK dilution plate (the cold stage). Similarly, during ground tests, a ~ 30 nW heat load was observed on the bolometer plate (the detector stage) when operating the cooler.⁹ These extra heat loads were attributed to the dissipation of the compressor microvibrations and were essentially a constant heat load. The result was a non-negligible drop in available cooling power by ~ 55 nW due to the dissipation of microvibrations, from about 200 nW to 145 nW. To my knowledge, this was the first observation and measurement of the thermal impact of microvibrations in space.

Indeed, the coldest cooling stage of sub-Kelvin instruments (typically an ADR or dilution refrigerator), is often only able to provide at most a few microwatts of cooling power. Several tens of nanowatts of stray heat load due to dissipated microvibrations at the coldest stage are therefore non-negligible.

The impact of a fluctuating heat load due to the vibration of cryocoolers was observed on the calorimeter of the engineering model of the Astro-H/SXS instrument. When on, the cryocoolers caused an increase in temperature (corresponding to a constant heat load of the order of 10 nW), though more importantly also an increase in its fluctuation.¹⁵ The temperature jump is visible left side of Figure 1.5, while the jump in the temperature fluctuation is shown on the right side. It is seen to jump from around 2 μ K to over 40 μ K, degrading the energy resolution of the microcalorimeter.

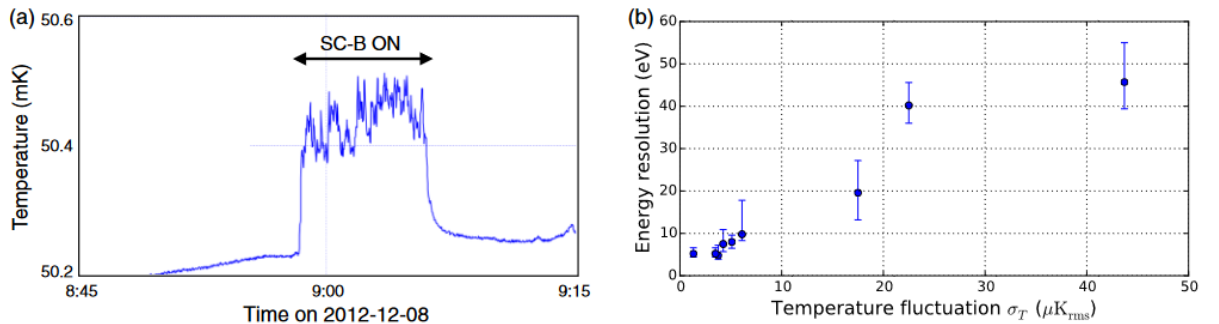


Figure 1.5 – Cryocooler microvibrations impact on temperature (left) and energy resolution (right) of Astro-H/SXS microcalorimeter¹⁵

¹⁵ (Takei, Yasuda, Ishimura, & Iwata, 2018)

Bearing in mind that the thermal requirements for the microcalorimeter were to maintain a temperature of 50 mK with a stability of $2 \mu\text{K}^{11}$, the temperature fluctuations induced by dissipated microvibrations therefore far exceeded said requirements.

It is therefore crucial to minimize both the constant heat load of dissipated microvibrations (as in the case of Planck) and the fluctuations of this heat load (as in the case of Astro-H) to the thermal detector. These thermal impacts are similar yet distinct. The lessons learned from such past missions, highlight the importance of understanding how microvibrations could dissipate in the form of heat and affect the performance of cryogenic detectors. This topic is particularly critical for the future Athena mission, being cryogen-free (thus increased microvibration disturbance from cryocoolers) and seeking higher energy resolution for the X-IFU instrument than its Japanese predecessor Astro-H/SXS.

The serious degradation of the detector performance of Astro-H also made microvibration disturbance a primary concern in the development of XRISM. A comprehensive study of the mechanical signature of its microcalorimeter was performed by JAXA, who found harmonics of the Stirling and JT coolers.¹⁶ System-level tests also studied the influence of RWs and inertial reference units (IRU). Thankfully, it was concluded for XRISM that the excess heat load (measured by the ADR current consumption rate) and temperature stability (measured by the fluctuations of the 50 mK stage) were sufficiently low to not threaten the required performance of its microcalorimeter.¹⁷

To ensure the performance of ever more sensitive instruments, a deeper understanding of microvibration dissipation in cryogenic thermal detectors is therefore needed. This thesis focuses on said dissipation at the coldest cooling stage rather than the detector stage, as well as in common elementary cryogenic devices. Looking ahead, it is now worth covering relevant research already carried out in this area.

1.2 Additional literature

Literature in the field of microvibration dissipation in low temperature space applications is sparse. Here we will first discuss the only article in this field (to our knowledge) in which dedicated microvibration measurements are presented. This article was presented at CEC in 2019 and published just before my PhD in 2020. We then discuss the takeaways of two other articles that broaden our approach by offering useful insights into the influence of temperature on material damping.

1.2.1 Observation of microvibration dissipation in Kevlar fibres

ADR stages, usually the coldest stage of a cryogenic chain, are typically insulated using Kevlar. This synthetic fibre displays high insulation without compromising mechanical strength and stiffness. Both adequate insulation and structural soundness can be guaranteed using a Kevlar suspension system, where Kevlar strings are routed and tensioned in such a way so as to suspend the ADR cold stage.

¹⁶ (Ryuta, et al., 2022)

¹⁷ (Takashi, et al., 2022)

As discussed, the coldest cooling stages of sub-Kelvin instruments often only provide a few microwatts of cooling power. Stray dissipations near these stages, even if minor, could reduce the cooling power or performance of the detectors they cool.

During lab testing, NASA GSFC observed significant heat loads on the coldest ADR stages, thought to be due to high vibration levels originating from a mechanical cryocooler.¹⁸ This intermittent heat load exceeded 10 μ W in some instances, consuming the entirety of the available cooling power and making continuous 50 mK temperatures unattainable. It was theorized that this heat load could be from the plastic deformation of, and friction between, Kevlar fibres. Subsequent tests were performed to confirm this.

A bare-chip thermometer was placed on 1 of the 6 Kevlar legs of an insulated dummy mass (mimicking an ADR cold stage), as shown in the schematic of Figure 1.6. Its temperature was regulated slightly above that of the support frame. The temperature elevations in the Kevlar string were calibrated to the cold stage regulation power drop, the latter a direct indication of the additional heat load there. Heat load at the Kevlar string and on the cold stage could then be compared.

When the mechanical cryocooler was powered off, though massively shortening the ADR run time, said excess heat load vanished. Furthermore, using a transducer, a resonance at 128 Hz (measured using an interferometric sensor) was found to heat up both the Kevlar string and the cold stage itself. This suggests that microvibration dissipation occurred, at least in part, in the Kevlar strings.

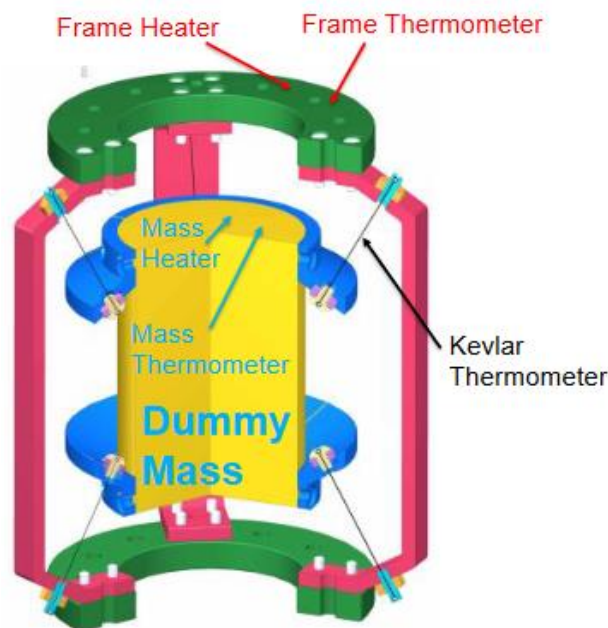


Figure 1.6 – Schematic of dummy mass suspended by Kevlar strings (one of which instrumented with a thermometer)¹⁸

This work was a pragmatic study of microvibration dissipation. The susceptibility of a test object (the dummy mass insulated by Kevlar strings) to a microvibration source (the cryocooler) was evaluated. A transducer was used to begin exploring mechanical excitation frequencies that cause dissipation. In our work however, we want to use a more fundamental approach. We are interested in the core mechanisms involved in microvibration dissipation, its location and the effect of varying mechanical excitation levels.

¹⁸ (Tuttle, et al., 2020)

Indeed most studies in our field of interest reflect an approach of microvibration mitigation rather than a wish for a more fundamental understanding of the phenomena at play.

The tests presented here nevertheless draw our attention to the difficulties of microvibration dissipation measurements at cryogenic temperatures. The reliability and repeatability of sub-microwatt dissipation measurement, as well as low level displacement and acceleration measurement, are particularly challenging. Examples of the work presented here include changes in Kevlar stiffness due to vacuum and temperature and uncontrolled dissipation of the sensor laser beams on the cold stage. For our own work, we must therefore be aware of the sensitivity of our setup to our test environment and the sensing equipment we use, as these will impact the quality of the fine measurements we wish to make.

1.2.2 Impact of low temperature on mechanical properties

Temperature generally has an impact on the mechanical properties of materials. This directly affects microvibration dissipation that may exist in a system we study. A recurring topic throughout this thesis, it is therefore key that we understand the effect of low temperature on mechanical behaviour. Regarding literature in this area, two articles in particular stand out.

The first outlines experiments carried out at NASA's JPL (Jet Propulsion Laboratory)¹⁹, where the damping of various materials (metallic, composite and ceramic) was studied at cryogenic temperatures (down to 20 K). Viscous damping (where the resistive force acting on an object is proportional to its velocity) was found to decrease at lower temperatures in the test specimens (rectangular bars subjected to bending strain), with each material showing a different rate of decrease. This evolution is shown for different Aluminium samples that differ only in geometry in Figure 1.7.

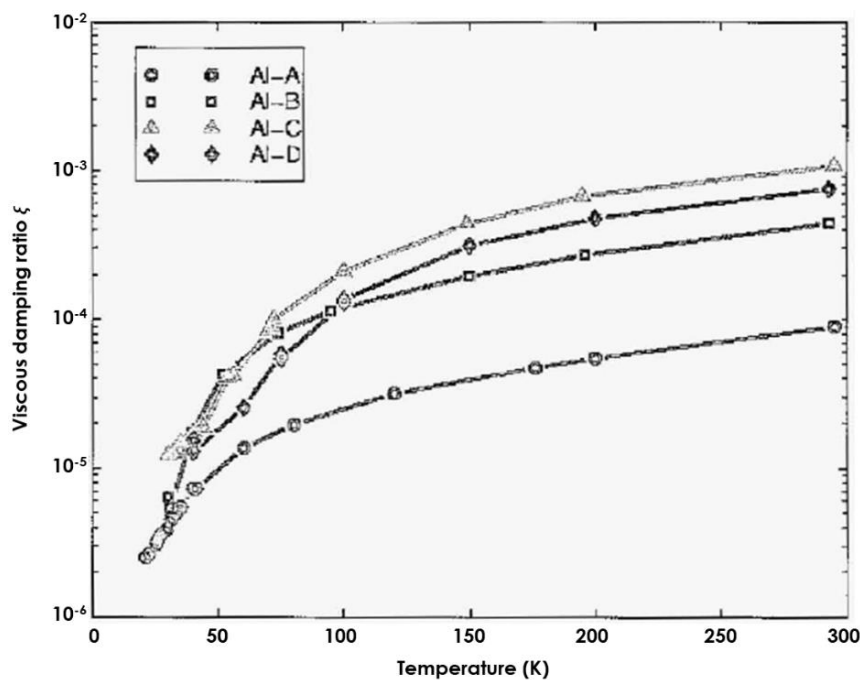


Figure 1.7 – Evolution of viscous damping ratio in Aluminium samples cooled down to 20 K¹⁹

¹⁹ (Levine & White, 2003)

The most sensitive were the metallic materials studied, damping decreasing from 293 K to 40 K up to a factor 61 for Aluminium and a factor 12 for Beryllium. This corresponds to a large increase of the quality factor, a quantity that reflects the damping present in an oscillating system (discussed later in this chapter).

Similar studies were carried out at the Max-Planck institute²⁰, where the impact of low temperature on the mechanical resonance of bars of various Aluminium alloys was studied. Between 295 K and 4.5 K their resonance frequency increased by up to ~8% and the quality factor by up to a factor ~220. The increase in the Q factors by ~2 orders of magnitude of the different Aluminium alloy bars is visible in Figure 1.8. The concept of the Q factor is introduced and discussed later in this chapter.

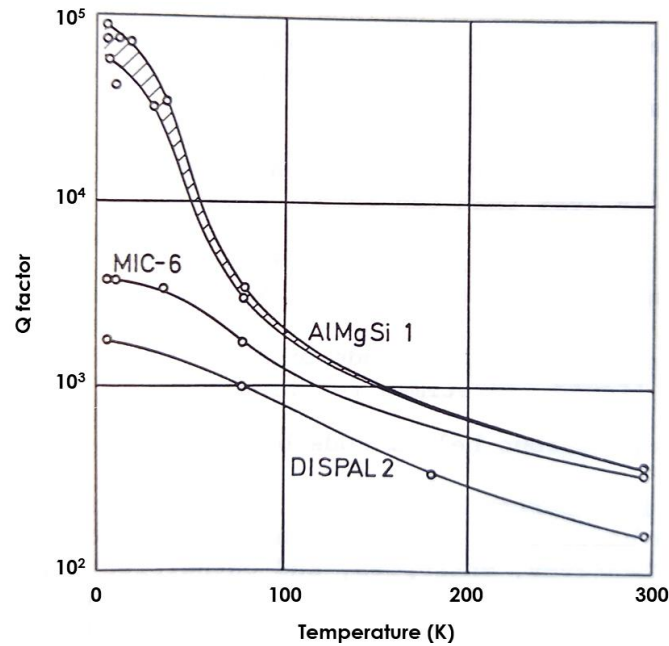


Figure 1.8 – Temperature dependence of Q factor for 3 Aluminium alloys²⁰

This literature highlights a high sensitivity of mechanical behaviour to cryogenic temperatures, in particular of metallic materials, whose properties are more susceptible to temperature than certain composites or ceramics. The above examples also direct our expectations regarding the evolution of mechanical damping and the Q factor when cooling to cryogenic temperatures.

1.3 Damped mass-spring model

During this thesis, the study of dissipation due to microvibrations at cryogenic temperatures has primarily been experiment-based. Analytical models can strengthen or refute hypotheses derived from such experiments. Models may themselves also shed light on yet unexplored experimental phenomena. This complementarity has motivated the development of a simple thermomechanical dissipation model.

This model aims to estimate heat dissipation induced by microvibrations. The forced harmonic excitation of a damped mass-spring system forms its basis. Though it is a well-studied classical mechanical system, we outline it in detail as it is central to our work. In particular, we use it to bridge the gap between the mechanical excitation and the heat dissipation it induces. The source of this dissipation is assumed to be

²⁰ (Katterloher, 1988)

the work carried out by the viscous force (i.e. proportional to the velocity). Having such a model at my disposal supports the interpretation and analysis of my experimental results.

1.3.1 Kelvin-Voigt model

The simplest description of a mechanically oscillating system is simple harmonic motion (SHM). The 1 degree-of-freedom (DOF) motion of a mobile mass attached to a spring exerts a restoring force proportional to the distance of the mass from an equilibrium position. Given an initial displacement from this position, the mass will oscillate indefinitely in the absence of energy dissipation.

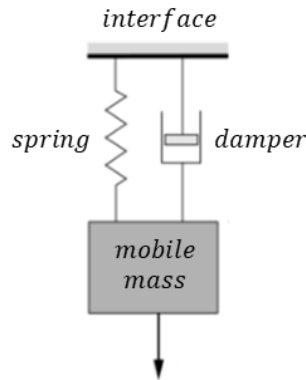


Figure 1.9 – Damped 1-DOF mass-spring system

All real-world systems though, however efficient, are dissipative and thermodynamically irreversible. In the case of oscillating systems, amplitudes of motion will decay over time due to the dissipation of energy. Extending SHM to include damping is a necessary step in attempting to model microvibration-induced dissipation. We do this by introducing the mechanical damper shown in Figure 1.9.

This is akin to the simplest viscoelastic model of a material: the Kelvin-Voigt model.²¹ It is shown in Figure 1.10: a purely elastic spring (described by the Young modulus E) in parallel with a purely viscous damper (described by a damping coefficient R). Due to an external stress σ , the material experiences a strain ϵ .

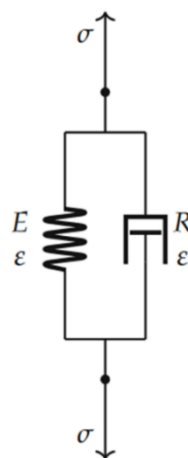


Figure 1.10 – Representation of Kelvin-Voigt model²¹

²¹ (Gaul & Schmidt, 2022)

In the Kelvin-Voigt model, the phase shift between the strain ε (the *displacement* of our mass-spring system) and the stress σ (the *force* exerted by the viscous damper) allows us to define an energy dissipation. Specifically, it is given by the surface area of the hysteresis loop between these two quantities, an example of which is shown in Figure 1.11. Section 1.3.5.4 will detail how energy dissipation is calculated in our damped mass-spring system from the work carried out by the viscous damping force.

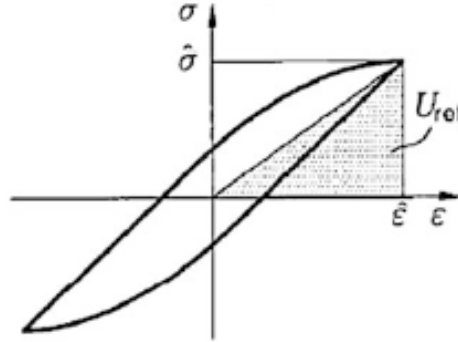


Figure 1.11 – Hysteresis loop between stress and strain in Kelvin-Voigt model²¹

1.3.2 Damping and its forms

Damping in mechanical systems occurs in various forms, the most common being discussed below.

Coulomb (or dry friction) damping dissipates energy via friction, such as between two sliding surfaces in contact. It is proportional to the normal force between them and is independent of relative displacement or velocity. It always acts in opposition to the direction of motion and being of constant value (i.e. a fixed energy loss per cycle), it decays linearly over time. Representing damping using a constant value though, fails to reflect most real-world scenarios that decay exponentially.

$$F_{coulomb} = -A \quad (1.1)$$

Solid (or hysteretic or material) damping occurs due to the deformation of a solid. Considered as internal friction, it is dissipated as heat. On a stress-strain curve, the bending and unbending of a solid will give rise to a hysteretic loop, the area of which representing the energy dissipated. This type of damping is generally proportional to deformation (i.e. a state of higher stress and higher strain). Solid damping is predominant in rubber elastic bands for example.

$$F_{solid} = -Ax \quad (1.2)$$

Viscous (or fluid) damping also opposes motion but is proportional to velocity rather than displacement, meaning faster motions dissipate more than slower ones with this form of damping. Viscous damping is typically the predominant form of damping in vehicle shock absorbers for example.

$$F_{viscous} = -A\dot{x} \quad (1.3)$$

The overall damping in most mechanical systems will often be a combination of several forms of damping. However, the most common modelling choice assumes viscous damping. For this reason, the damped mass-spring model presented from section 1.3.4 onwards also only considers viscous damping.

1.3.3 Hypotheses and boundary conditions

Modelling dissipation by viscous damping is the most significant hypothesis of this model. This form of damping assumes the dissipative force is proportional to the velocity of the mobile mass.

This modelling choice simplifies the solution of the second-order differential equation involved. We further assume that this damping occurs due to the relative motion between the interface and mobile mass, and not due to the absolute motion of the mobile mass (even if we find that these are close).

Another hypothesis made is that of 1-DOF behaviour. In reality, all systems portray multi-DOF behaviour but some are *predominantly* 1-DOF (pendulums, doors, etc.). Despite real-world systems never being purely 1-DOF, the measurement procedures later discussed allow for this hypothesis.

We also hypothesise constant system stiffness (i.e. that Hooke's law is valid). Within the elastic range of materials, the linear relationship between stress and strain is generally true. However when exceeding the elastic limit, the spring stiffness is not constant and becomes a function of strain (or displacement). Thus, the equivalent hypothesis made is that the material elastic limits are not exceeded.

Regarding boundary conditions in the context of our study on microvibrations, the mobile mass can only be excited through its mechanical interface. Displacement and force boundary conditions would be valid modelling choices. The displacement boundary condition is chosen however (and later developed in the model), as it reflects the capabilities and limits of the experimental setup. We will see that a displacement boundary condition can be defined thanks to cryogenic accelerometers mounted near the interface.

1.3.4 Model outline

To describe the behaviour of the damped mass-spring system we define system parameters m , k , and c , respectively the mobile mass [kg], spring stiffness [N/m] and damping coefficient [Ns/m]. Note that the damping coefficient is not to be confused with the damping ratio (denoted below ξ).

The displacement of the mobile mass is denoted x . The displacement of the interface (i.e. the forced oscillatory displacement representing induced microvibrations) is denoted y . These system parameters are summarised in Figure 1.12.

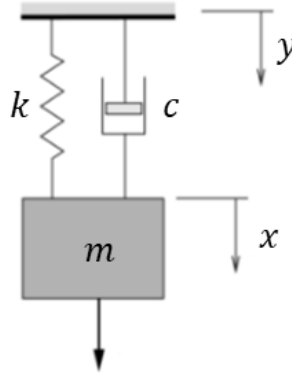


Figure 1.12 – Damped 1-DOF mass-spring system with parameters

The forced harmonic motion of said damped system is described by:

$$m\ddot{x} + c(\dot{x} - \dot{y}) + k(x - y) = 0 \quad (1.4)$$

The forced displacement y at an imposed frequency ω , and amplitude Y_0 , is defined by:

$$y(t) = Y_0 \sin(\omega t) \quad (1.5)$$

Prior to developing the solution to (1.4), we must first cover key concepts: undamped and damped resonance, the damping ratio and the Q factor.

1.3.4.1 Resonance

For the system outlined, the undamped angular resonance (or natural) frequency ω_0 and damped angular resonance frequency ω_d are given by:

$$\omega_0 = \sqrt{\frac{k}{m}} \quad (1.6)$$

$$\omega_d = \sqrt{\frac{k}{m} - \left(\frac{c}{2m}\right)^2} \quad (1.7)$$

We notice that as damping (represented by the damping coefficient c) increases, the system's damped natural frequency tends to decrease. For low enough damping, particularly in underdamped systems as discussed later, the undamped and damped natural frequencies are very close.

1.3.4.2 Damping ratio and Q factor

A critical damping coefficient c_c can also be defined above which the system does not resonate (i.e. it is overdamped), namely when the radical of (1.7) cancels out:

$$\frac{k}{m} - \left(\frac{c_c}{2m}\right)^2 = 0 \quad (1.8)$$

and so

$$c_c = 2\sqrt{mk} = 2m\omega_0 \quad (1.9)$$

The *damping ratio* is defined as the ratio of the damping coefficient c to the critical damping c_c :

$$\boxed{\xi = \frac{c}{2m\omega_0}} \quad (1.10)$$

or solely in terms of parameters m , k and c :

$$\boxed{\xi = \frac{c}{2\sqrt{mk}}} \quad (1.11)$$

The dimensionless parameter known as the Q (or “*quality*”) factor describes how damped an oscillating system is. The concept of Q is suitable for the study of the impact of microvibration in an oscillating damped mass-spring system. In terms of the damping ratio, the Q factor is defined as:

$$Q = \frac{1}{2\xi} \quad (1.12)$$

For the system outlined, the Q factor in terms of system parameters is therefore:

$$\boxed{Q = \frac{m\omega_0}{c}} \quad (1.13)$$

The Q factor is discussed further in section 1.3.7, after the detailed solution and analysis of the model.

1.3.4.3 Damping cases

Recalling expression (1.12), giving the Q factor in terms of the damping ratio, we can define the three situations for the damped mass-spring system:

- $\xi > 1 \equiv Q < \frac{1}{2}$ → *overdamped* (no system resonance)
- $\xi = 1 \equiv Q = \frac{1}{2}$ → *critically damped* (no system resonance)
- $\xi < 1 \equiv Q > \frac{1}{2}$ → *underdamped* (system resonance)

Our model focuses on underdamped cases, for Q values above 100. This is because our test campaigns have always involved underdamped systems showing Q factors larger than this, and dissipation due to microvibrations close to their mechanical resonance.

1.3.5 Model solution

The resulting harmonic motion x of the mobile mass will be composed of the homogeneous solution x_h (transient response) and particular solution x_p (steady-state response):

$$x(t) = x_h(t) + x_p(t) \quad (1.14)$$

As $t \rightarrow \infty$ the transient response will decay and ultimately vanish, the steady-state response then fully describing the motion of the mobile mass under a forced stable regime. The forced steady state regime is of greater interest in the estimation of heat dissipation induced by microvibrations. For this reason, only the solution for the steady state is detailed. The solution of the transient responses is simply given.

1.3.5.1 Steady state response

The steady state response, given by the particular solution, must satisfy the forced harmonic motion described by (1.4). It can be rewritten as:

$$m\ddot{x}_p + c\dot{x}_p + kx_p = c\dot{y} + ky \quad (1.15)$$

By the definition of y this gives:

$$m\ddot{x}_p + c\dot{x}_p + kx_p = c\omega Y_0 \cos(\omega t) + kY_0 \sin(\omega t) \quad (1.16)$$

$$= F_0 \sin(\omega t - \phi) \quad (1.17)$$

Where

$$F_0 = Y_0 \sqrt{(c\omega)^2 + k^2} \quad (1.18)$$

$$\cos \phi = \frac{c\omega Y_0}{F_0} \quad \sin \phi = \frac{kY_0}{F_0} \quad (1.19)$$

For simplicity of the solution, we assume $\phi = 0$. This removes the otherwise induced shift in the particular solution for the case where $\phi \neq 0$, given our focus here on the *amplitudes* of motion.

Thus the particular solution x_p must satisfy:

$$m\ddot{x}_p + c\dot{x}_p + kx_p = F_0 \sin(\omega t) \quad (1.20)$$

The solution is assumed to be of the form:

$$x_p(t) = X \sin(\omega t - \alpha) \quad (1.21)$$

Substituting (1.21) into (1.20) leads to:

$$\begin{aligned} m[-X\omega \sin(\omega t - \alpha)] + c[X\omega \cos(\omega t - \alpha)] + k[X \sin(\omega t - \alpha)] \\ = F_0 \sin(\omega t) \end{aligned} \quad (1.22)$$

Through expansion of the $\sin(\omega t - \alpha)$ and $\cos(\omega t - \alpha)$ terms using trigonometric identities, regrouping of the resulting $\sin(\omega t)$ and $\cos(\omega t)$ terms and comparison of both sides of the final equation, one obtains:

$$X[(k - m\omega^2) \cos \alpha + c\omega \sin \alpha] = F_0 \quad (1.23)$$

$$X[(k - m\omega^2) \sin \alpha - c\omega \cos \alpha] = 0 \quad (1.24)$$

Solving for X and α , and substituting (1.18) yields:

$$\alpha = \tan^{-1} \left(\frac{c\omega}{k - m\omega^2} \right) \quad (1.25)$$

$$X = Y_0 \sqrt{\frac{(c\omega)^2 + k^2}{(c\omega)^2 + (k - m\omega^2)^2}} \quad (1.26)$$

Substituting the expressions of ω_0 , ω_d and Q , namely (1.6), (1.7) and (1.13), into (1.25) and (1.26) gives:

$$\alpha = \tan^{-1} \left[\frac{\omega_0 \omega}{Q(\omega_0^2 - \omega^2)} \right] \quad (1.27)$$

and

$$X = Y_0 \sqrt{\frac{\left(\frac{\omega_0 \omega}{Q}\right)^2 + \omega_0^4}{\left(\frac{\omega_0 \omega}{Q}\right)^2 + (\omega_0^2 - \omega^2)^2}} \quad (1.28)$$

Finally, the steady state response of an underdamped mass-spring system can be written as:

$$x_p(t) = Y_0 \sqrt{\frac{\left(\frac{\omega_0 \omega}{Q}\right)^2 + \omega_0^4}{\left(\frac{\omega_0 \omega}{Q}\right)^2 + (\omega_0^2 - \omega^2)^2}} \sin(\omega t - \alpha) \quad (1.29)$$

We notice that the long-term motion of the damped mass-spring system is harmonic motion at the imposed frequency ω of the interface. The response also lags the imposed motion y by an angle α .

1.3.5.2 Transient response

The transient response of an underdamped ($\xi < 1$) mass-spring system is of the form:

$$x_h(t) = A_h e^{-\xi \omega_0 t} \sin(\omega_d t - \theta) \quad (1.30)$$

We note an oscillation at the damped natural frequency ω_d when the system is allowed to oscillate freely. The amplitude of these oscillations decays exponentially, with a time constant equal to $1/\xi \omega_0$. For resonances of several hundred Hertz such as for the systems studied in this thesis, the transient response is therefore very short. We therefore only focus on the steady-state regime.

1.3.5.3 Amplification

The output displacement amplitude X of the steady state response given by equation (1.28) varies with ω and Q . It is interesting to study the evolution of the mechanical amplification $A_{mech} = X/Y_0$ for excitation frequencies near ω_0 , and for different Q values. Equation (1.28) can be rewritten as:

$$A_{mech} = \frac{X}{Y_0} = \sqrt{\frac{\frac{1}{Q^2} \left(\frac{\omega}{\omega_0}\right)^2 + 1}{\left(\frac{\omega}{\omega_0}\right)^4 + \left(\frac{1}{Q^2} - 2\right) \left(\frac{\omega}{\omega_0}\right)^2 + 1}} \quad (1.31)$$

We plot, for different Q values, mechanical amplification normalised to the Q factor $(X/Y_0)/Q$ versus the excitation frequency normalised to the undamped resonance ω/ω_0 . The result is shown in Figure 1.13, for different low Q factor values (though still underdamped cases). The red crosses show their peaks.

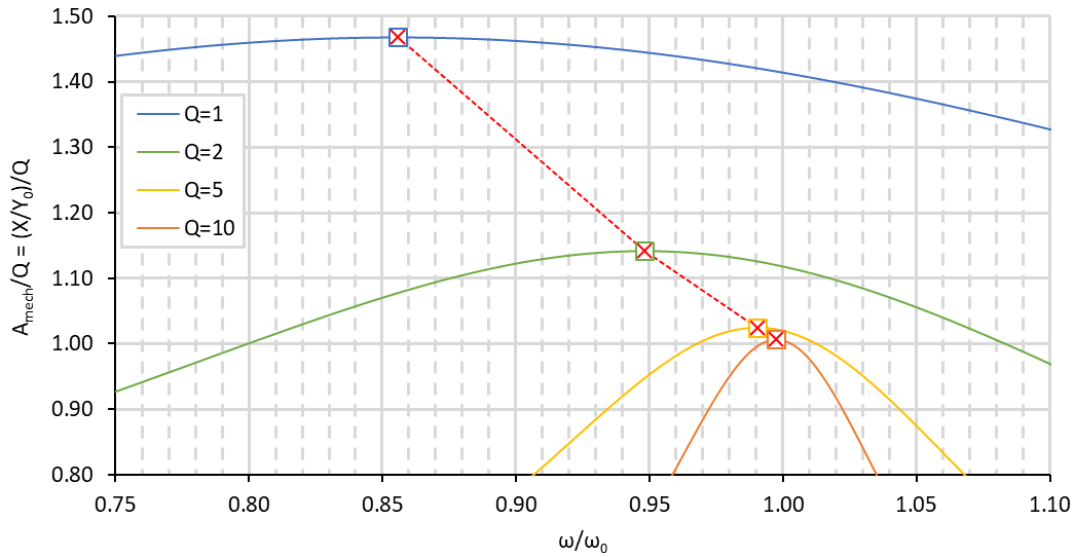


Figure 1.13 – Mechanical amplification (normalised to Q) versus angular excitation frequency (normalised to undamped resonance), for different low Q factor values (but still underdamped cases)

We observe that as the Q factor increases (i.e. less damping), peak amplification occurs at a frequency that fast approaches the undamped natural frequency, which is expected. This normalised peak amplification also rapidly approaches 1, that is to say amplification approaches Q .

Table 1.1 shows for the 4 values of Figure 1.13 ($Q = 1, 2, 5$ and 10) and 2 additional ones ($Q=100$ and 1000), the difference between the undamped natural frequency ω_0 and the peak amplification frequency of the forced regime ω_{max} . We set $f_0 = 200$ Hz, representing a resonance frequency close to the main test object studied in this thesis.

Table 1.1 – Effect of Q on peak amplification frequency

Q	$\Delta f (= f_0 - f_{max})$
1	≈ 29 Hz
2	≈ 10 Hz
5	≈ 2 Hz
10	≈ 0.5 Hz
100	≈ 5 mHz
1000	≈ 40 μ Hz

For just $Q > 5$, the difference is less than 1%. For $Q > 100$ this falls to less than 0.0025%. As discussed, our model only considers highly underdamped cases, for Q factors exceeding 100. Hence for our application, we can state that:

$$\omega_{max} = \omega_0 \quad (1.32)$$

And the magnitude of this amplification is:

$$A_{mech} = Q \quad (1.33)$$

1.3.5.4 Dissipation

An analytical expression for dissipation induced by microvibrations could offer insights into experimental observations and the underlying dissipation phenomena. For the viscous damped mass-spring system presented, two methods were investigated that lead to the same expression of the power dissipation.

The first estimates the energy injected to the system from the work done by the interface. In the steady state the average energy over one cycle is constant, hence any energy injected must equal the energy dissipated over one cycle. Scaled to the excitation frequency, we then obtain average power dissipation.

The second method deduces the dissipation from the work done by the viscous force, per cycle. As with the first method, the average power dissipation is then the energy dissipated per cycle scaled to the excitation frequency. Only this method is developed further given that both lead to the same result.

In the highly underdamped case such that Q is very large, we can assume $\dot{x} \gg \dot{y}$. The viscous force, opposing the direction of motion and proportional to the difference in velocity (see section 0) becomes:

$$F_{visc} = -c\dot{x} \quad (1.34)$$

The power dissipation, that is the work done by this force, is thus:

$$P_{diss}(\omega) = \frac{1}{T} W_{visc/cycle} = \frac{1}{T} \int_{x(t_0)}^{x(t_0+T)} F_{visc} dx = \frac{1}{T} \int_{x(t_0)}^{x(t_0+T)} c\dot{x} dx \quad (1.35)$$

Given the first derivative of the displacement x :

$$\frac{dx}{dt} = \omega X \cos(\omega t - \alpha) \quad (1.36)$$

We can change the variables of the integral of (1.35):

$$P_{diss}(\omega) = \frac{1}{T} \int_0^T c \omega^2 X^2 \cos^2(\omega t - \alpha) dt \quad (1.37)$$

Apart from the cosine function, all parameters are independent of time:

$$P_{diss}(\omega) = c \omega^2 X^2 \cdot \frac{1}{T} \int_0^T \cos^2(\omega t - \alpha) dt = \frac{c \omega^2 X^2}{2} \quad (1.38)$$

Substituting (1.13) to replace the damping coefficient c , we obtain the time-average power dissipation for a given excitation frequency ω :

$$P_{diss}(\omega) = \frac{m \omega_0}{2Q} \omega^2 X^2 \quad (1.39)$$

1.3.6 Model analysis

Given the model solution just developed its behaviour can be studied for different situations.

1.3.6.1 Maximum displacement

The mechanical amplification discussed in section 1.3.5.3 corresponds to the amplification of all mechanical quantities. We can therefore state that:

$$X_{max} = Q Y_0 \quad (1.40)$$

1.3.6.2 Maximum acceleration

As a boundary condition, it is of interest to introduce the amplitude of the interface acceleration a , deduced from the second derivative of the interface displacement y defined in section 1.3.4:

$$a = Y_0 \omega^2 \quad (1.41)$$

We note that a is the *injected* interface acceleration (i.e. at interface level), not the output acceleration $\ddot{X}$ of the mobile mass. The mechanical amplification discussed in section 1.3.5.3 applies here too:

$$\ddot{X}_{max} = Q a \quad (1.42)$$

1.3.6.3 Maximum dissipation

As per (1.39), given that the average power dissipation for a given excitation frequency ω is proportional to X^2 and that all other parameters are constant, it will cross its maximum when X^2 also does. As we have seen, this occurs at $\omega \approx \omega_0$ for Q sufficiently large. Substituting into (1.39) gives:

$$P_{max} = [P_{diss}(\omega)]_{max} = \frac{m\omega_0^3}{2Q} X_{max}^2 \quad (1.43)$$

Next, we substitute the expression for X_{max} , giving:

$$P_{max} = [P_{diss}(\omega)]_{max} = \frac{mQY_0^2\omega_0^3}{2} \quad (1.44)$$

Finally, it is useful to express P_{max} in terms of measurable input parameters. By substituting the expression for the interface acceleration a , we obtain the maximum time-average power dissipation of the viscous damped mass-spring system:

$$P_{max} = \frac{mQ}{2\omega_0} a^2 \quad (1.45)$$

We note a proportionality between the maximum power dissipation and the square of the amplitude of the injected interface acceleration.

$$P_{diss} \propto a^2 \quad (1.46)$$

This proportionality stems from the proportionality with X^2 , the amplitude of the injected interface displacement, as per (1.39). Furthermore, the proportionality is true not only at resonance but at all excitation frequencies. It is later compared to true dissipations observed experimentally.

1.3.6.4 Mechanical profiles

Given the expression of X (1.28) and a set of parameters and initial conditions (IC), the amplitude profiles of the displacement, speed and acceleration of the mobile mass can be plotted against the excitation frequency ω .

By derivation, the amplitudes of the displacement, speed and acceleration are respectively X , ωX and $\omega^2 X$. The set of parameters and initial conditions chosen, based broadly on experimental values, is shown in Table 1.2.

Table 1.2 – Set of parameters and initial conditions (reference case)

	Parameter / IC	Value
quality factor	Q	500
mobile mass	m	19.0 g
undamped natural frequency	$f_0 = \frac{\omega_0}{2\pi}$	220 Hz
interface acceleration at natural frequency (IC)	a	5 mg _{rms}

This set of parameters and initial conditions fully defines the system and its behaviour. The amplitudes of the displacement, speed and acceleration of the mobile mass are shown in Figure 1.14.

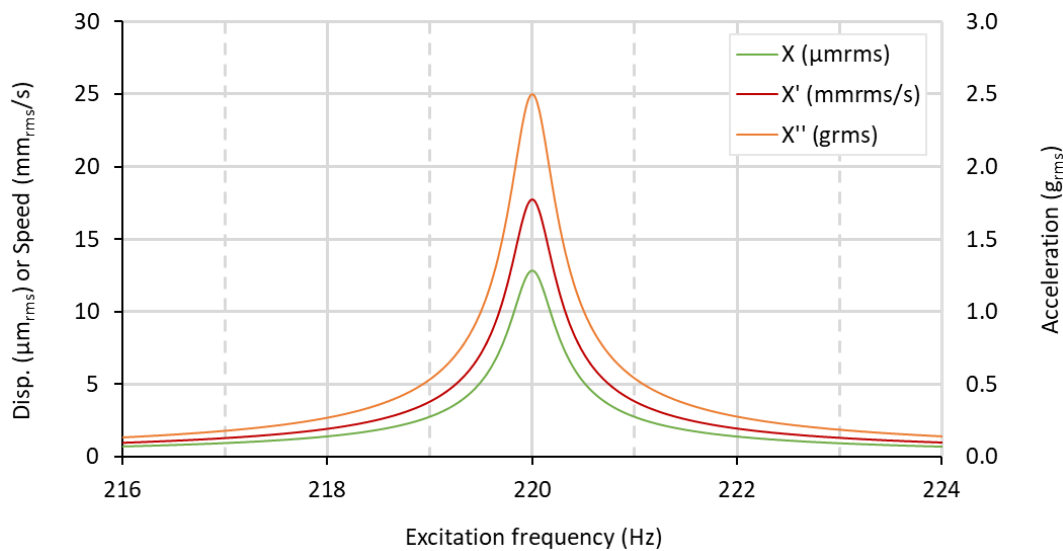


Figure 1.14 – Mechanical profiles of the mobile mass (reference case)

Table 1.3 summarises the maximum values of these curves. We note that at the natural frequency, the relationships between the peak values, namely $\dot{X}_{max} = \omega_0 X_{max}$ and $\ddot{X}_{max} = \omega_0^2 X_{max}$ hold true.

Table 1.3 – Mechanical profiles maxima (example case)

	Parameter	Maximum
displacement	X	13 μm _{rms}
speed	$\dot{X}$	18 mm _{rms} /s
acceleration	$\ddot{X}$	2.5 g _{rms}

1.3.6.5 Dissipation profiles

The amplitude profile of the dissipation occurring in the damped-mass spring system can be plotted against the excitation frequency ω , its expression given by (1.39). The same set of parameters and initial conditions is used as for the mechanical profiles (see Table 1.2). The resulting power dissipation response is shown in Figure 1.15.

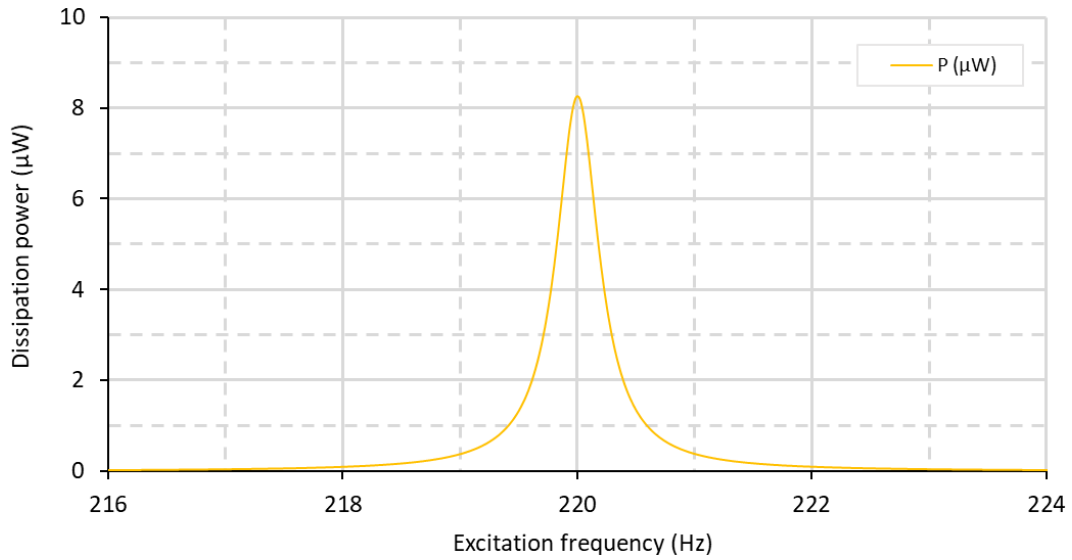


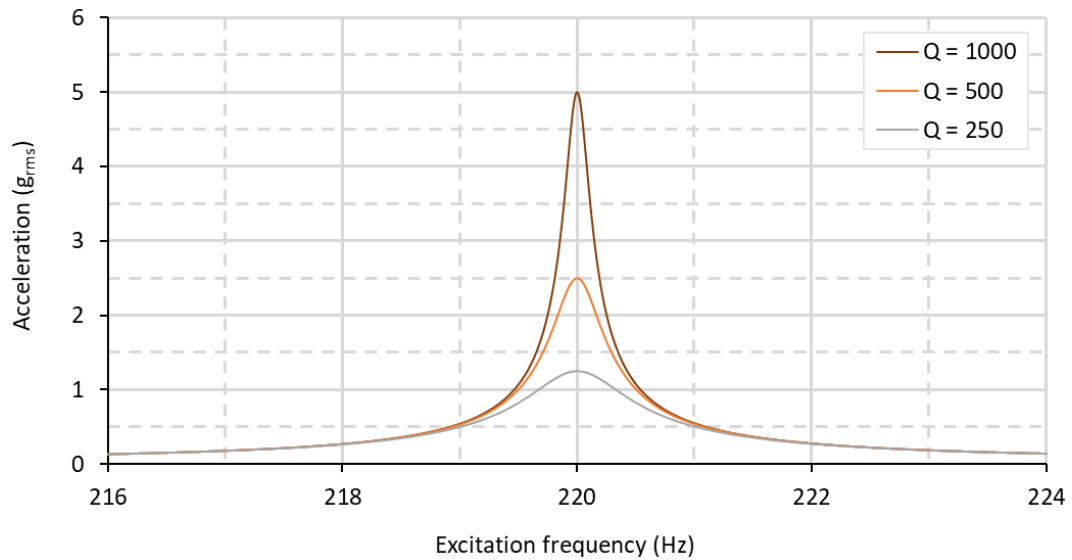
Figure 1.15 – Power dissipation profile (reference case)

We note that the maximum power dissipation ($\sim 8.3 \mu\text{W}$) corresponds to the expression of the analytical maximum power dissipation that can also directly be deduced from (1.45).

1.3.6.6 Parametric study

For the example case presented in section 1.3.6.4, we vary parameter values to observe their effect on the mechanical and dissipation responses. We consider the effects of changes in the Q factor and in the injected interface acceleration a , on the acceleration response $\ddot{X}$ and power dissipation response P_{diss} .

The effects on the acceleration response are shown in Figure 1.16 (variations of Q) and Figure 1.17 (variations of a). The previously discussed reference ($Q = 500$ and $a = 5 \text{ g}_{rms}$) is in orange.

Figure 1.16 – Acceleration response for varying Q values (example cases)

The Q value is halved ($Q = 250$) and doubled ($Q = 1000$). We note a corresponding halving and doubling respectively of the acceleration response: 1.25 g_{rms} and 5.00 g_{rms} (from 2.50 g_{rms}). This is expected.

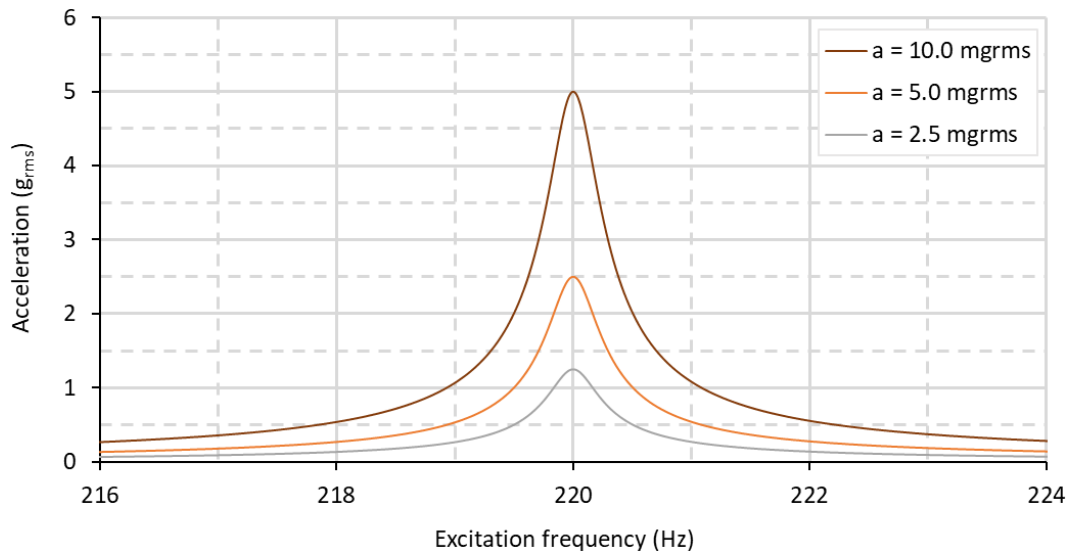
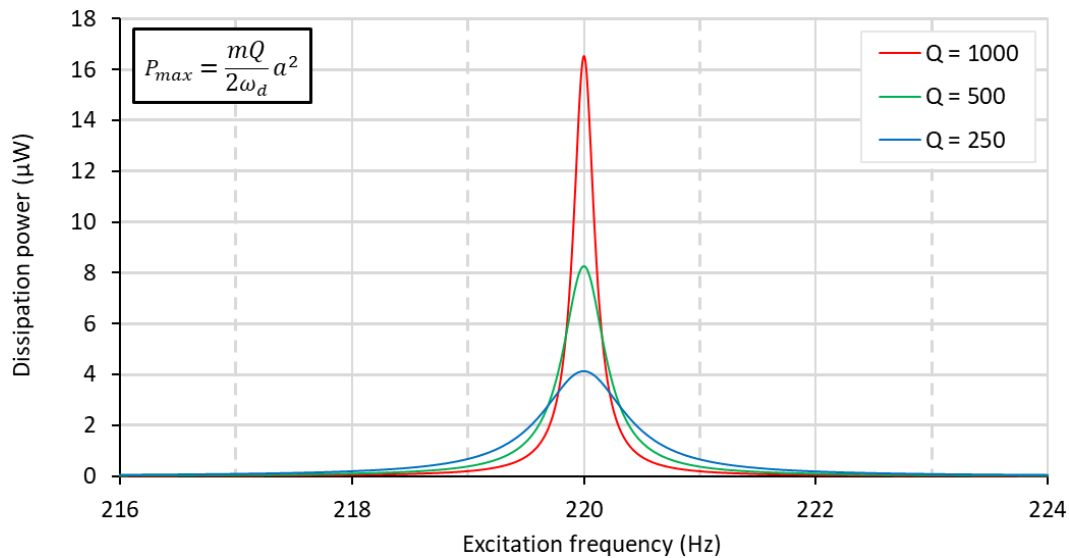


Figure 1.17 – Acceleration response for varying interface acceleration values (example cases)

This proportionality is also visible when varying a (for Q constant) and for the same reason. It too is halved ($a = 2.5 \text{ mg}_{\text{rms}}$) and doubled ($a = 10.0 \text{ mg}_{\text{rms}}$). The effects of varying Q or a differ in that at constant Q values (Figure 1.17) the bandwidth of the response is unchanged, whereas for varying Q values (Figure 1.16) the bandwidth changes: a higher Q value reduces the bandwidth. We refer to the bandwidth of the response as the spectral width at half maximum power (this is discussed further in section 1.3.7.1).

The effects on the power dissipation response are shown in Figure 1.18 (variations of Q) and Figure 1.19 (variations of a). The previously shown reference in Figure 1.15 is shown here in green.

Figure 1.18 – Dissipation response for varying Q values (example cases) and $a = 5 \text{ mg}_{\text{rms}}$

As with the acceleration response, we note a corresponding halving and doubling respectively of the dissipation response: $\sim 4 \text{ μW}$ and $\sim 16 \text{ μW}$ (from $\sim 8 \text{ μW}$). This too is expected.

Another interpretation we can make from equation (1.45) and that is visible in Figure 1.18 too, is that when damping is *lower* (a higher Q factor), peak dissipation power is in fact *higher*. We should keep this consequence in mind as it has major impacts for our experiments and can be counterintuitive.

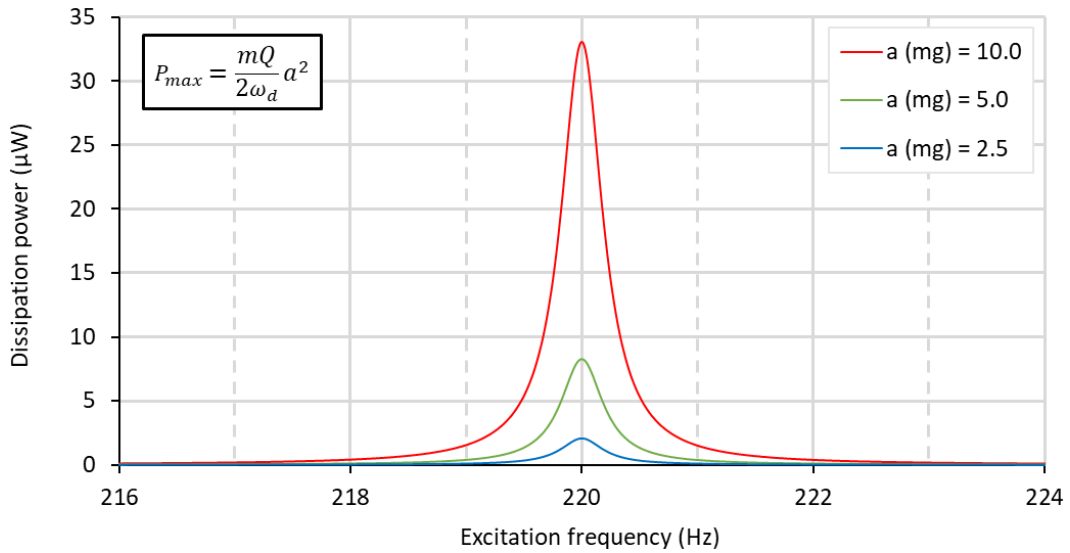


Figure 1.19 – Dissipation response for $Q = 500$ and varying interface acceleration levels (example case)

Contrary to the acceleration response, the power dissipation is not proportional to the interface acceleration a but rather to a^2 . In Figure 1.19 we therefore observe a fourfold decrease and increase due to the halving and doubling respectively of a : $\sim 2 \mu\text{W}$ and $\sim 32 \mu\text{W}$ (from $\sim 8 \mu\text{W}$).

1.3.7 Q factor discussion

1.3.7.1 Different definitions

The concept of the Q factor applies to dynamic systems in general. It has found applications in mechanics but also in material science and even in the study of celestial bodies (Earth's own “ Q factor” has been studied). In its broadest cross-disciplinary definition, the Q -factor is defined as the ratio of the energy stored by a resonator to the energy dissipated within it, per cycle. Formally, this “energy definition” is:

$$Q \stackrel{\text{def}}{=} 2\pi \frac{E_{\text{stored}}}{E_{\text{diss/cycle}}} \quad (1.47)$$

For large Q values and in the case of viscous-type damping for a mechanical resonator, an alternative but equivalent “bandwidth definition” also exists, based on the frequency-to-bandwidth ratio of the frequency response of an oscillating system. It is the ratio of the natural frequency to the full width at half-maximum (FWHM) power, corresponding to a level of approximately -3 dB:

$$Q \stackrel{\text{def}}{=} \frac{f_{\text{res}}}{\Delta f} \quad (1.48)$$

The FWHM may also be referred to as a signal's bandwidth. A high- Q resonator displays a high amplitude at its resonance but resonates only over a narrow frequency band (low FWHM). Likewise, a low- Q resonator displays a low amplitude at its resonance but resonates over a broad frequency band (high FWHM). This explains why in electronics, high- Q circuits are used in systems requiring high spectral

selectivity (e.g. radio receivers and superconducting RF cavities), as the amplitude response of such resonators falls rapidly when moving away from the natural frequency. High-Q systems therefore more easily distinguish between signals in the same spectral neighbourhood.

The bandwidth definition is used to estimate observed Q factors from both mechanical and thermal experimental measurements, without needing to estimate stored and dissipated energies.

In fact, we can only use the bandwidth definition for the dissipation measurements, as we do not have a “transfer function” of dissipation that would directly give the Q factor, as we can for mechanical amplification.

We should also note that the FWHM is directly the spectral width at half of the maximum for dissipation (in watts) but is the spectral width at half of the maximum *squared* for all mechanical quantities. This is because the bandwidth definition of the Q factor uses the width at half power not half amplitude.

1.3.7.2 Method validation

The model solution – for predicting both mechanical and dissipation behaviour – involves a single Q factor. Experimentally, Q is measured using the frequency-to-bandwidth ratio as discussed in section 1.3.7.1.

It is assumed that the “mechanical quality factor” Q_m and “thermal quality factor” Q_{th} are equivalent, and so comparable. In particular, it is worth verifying that the method of deducing the Q factor using the frequency-to-bandwidth ratio is indeed valid, whether for Q_m or Q_{th} .

The acceleration and dissipation response curves presented in section 1.3.6.4 and section 1.3.6.5 respectively are used as reference here. As per equation (1.48), the quality factor is deduced knowing the natural frequency (220 Hz) and the full-width at half-maximum (FWHM) power.

As discussed, the half-maximum *dissipation* power is simply half the peak of the dissipation response, its unit already being watts. The half-maximum *mechanical* powers however are proportional to the square of the mechanical responses. Hence the half-maximum powers are:

- Half-maximum power of *dissipation*: $\frac{P_{max}}{2}$
- Half-maximum power of *acceleration*: $\frac{\ddot{X}_{max}}{\sqrt{2}}$

Table 1.4 summarises the maxima and corresponding half powers in the respective units.

Table 1.4 – Half-maximum powers for mechanical and dissipation responses (example case)

	Parameter	Maximum power	Half-maximum power	Unit
<i>acceleration</i>	$\ddot{X}$	2.50	1.77	[g _{rms}]
<i>dissipation</i>	P	45.92	22.96	[μW]

Table 1.5 summarises for each response the numerically deduced natural frequency, FWHM's and the resulting frequency-to-bandwidth ratios that correspond to either Q_m or Q_{th} .

Table 1.5 – Quality factor estimations from frequency-to-bandwidth ratios (example case)

	<i>Parameter</i>	<i>Natural freq.</i>	<i>FWHM</i>	<i>Unit</i>	
<i>acceleration</i>	$\ddot{X}$	219.996	0.440	[Hz]	$Q_m = 499.84$
<i>dissipation</i>	P	219.996	0.440	[Hz]	$Q_{th} = 499.84$

The deduced Q factor values correspond to the imposed Q value of 500. The error in the deduced values ($\sim 0.032\%$) is simply due to the numerical discretisation. We conclude that the method of deducing the Q factor from the frequency-to-bandwidth ratio is valid.

We also note that caution must be taken as to the deduction of the FWHM. The signal from which it is calculated must be proportional to power. For our work, this corresponds to the dissipation profiles and the *square* of the mechanical profiles (displacement, velocity and acceleration).

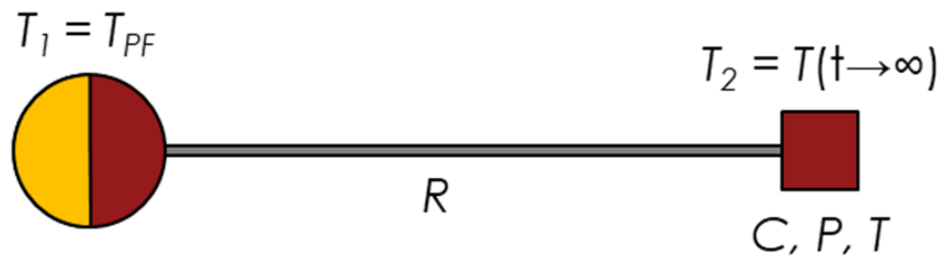
1.4 Generalised thermal model

Another model used to interpret thermal measurements and justify evolving the test configuration is the following thermal model. It is applicable to any test object studied in the manner presented in my thesis.

It is a simple thermal RC circuit in series. Here we will present it in its simplest form: a single pure thermal resistance R (the titanium tube of the heat switch) that insulates a single pure thermal capacitance C (the heat switch tip) at one of its ends. At its other end we impose a temperature T_1 (e.g. a cold source).

For no injected heat load the entire system is thermally homogeneous at temperature T_1 . When the capacitance is subjected to a heat load however, say from microvibration dissipation, its temperature will rise (given that the thermal resistance is not zero). Characterising the response of our system to such a heat load over time (i.e. its step response), helps us understand its impact and potentially quantify it.

For this, we additionally define the equilibrium temperature of the capacitance T_2 due to a heat load P arriving at the capacitance, such that $T_2 - T_1 = RP$. The parameters are shown in Figure 1.20.

Figure 1.20 – Thermal RC series circuit

The evolution of the temperature of the capacitance $T(t)$ due this heat load is given by:

$$T(t) = T_2 + (T_1 - T_2) e^{-t/\tau} \quad (1.49)$$

Where the thermal time constant τ [s] is the product of the capacitance C [J/K] and resistance R [K/W]:

$$\tau = R \cdot C \quad (1.50)$$

Hence the final temperature elevation $T_2 - T_1$ ($t \rightarrow \infty$) and the initial temperature slope ($t \rightarrow 0$) are:

- *Final temperature elevation [K]:* $\Delta T = RP$
- *Initial temperature slope [K/s]:* $\dot{T}(t \rightarrow 0) = \frac{P}{C}$

Hence, the final temperature elevation (i.e. thermal equilibrium) is independent of the capacitance C . Conversely, the initial temperature slope is independent of the thermal resistance R . As expected, we also note that both the final temperature elevation and initial slope are larger for a larger heat load P .

1.5 Thesis objectives

In this chapter, we emphasized the need of a deep understanding of microvibration and presented studies led at NASA to this purpose. We proposed two models that will help in the interpretation of results. Hence, our work will focus on building an experimental characterization with the following objectives:

1. Build a test bench that enables the measurement of sub-microwatt dissipation levels.
2. Build a test bench that enables the measurement of sub-mg acceleration levels.
3. Observe microvibration dissipation in common cryogenic devices at cryogenic temperatures.
4. Evaluate the amplitude of microvibration dissipation in common cryogenic devices.
5. Identify the mechanisms and location of microvibration dissipation.
6. Evaluate the validity of the damped mass-spring dissipation model by comparing its predictions to microvibration dissipation measurements.

This list outlines the broad objectives of this thesis, based largely on experimental work. Before presenting the main results in chapter 4, it is necessary to first introduce the test bench and detection chain in chapter 2, followed by the measurement environment and measurement methodologies in chapter 3.

Test bench and detection chain

The test bench can be broken down into its primary functions: (i) the cryogenic environment (ii) sensing and actuation equipment and (iii) voltage (i.e. sensor) measurement. The environment houses sensing and actuation equipment, whereas a separate test rack houses equipment for voltage measurement.

First, we shall present the full test bench, comprised of said environment, test rack and a lab PC. The test object and quantities studied will subsequently be covered, followed by the cryogenic environment (namely a helium bath cryostat). The mechanical and thermal equipment used for sensing and actuation will then be covered. To close this chapter, we will present the interfacing of all our equipment.

2.1 Introducing the test bench

The full test bench is represented in the schematic of Figure 2.1. The mechanical equipment is shown in blue, while the thermal equipment is shown in green. Double lines indicate communication between the lab PC, test rack and cryostat. Arrows are used for ease of comprehension: *commands* flow from left to right, whereas *readouts* flow from right to left.

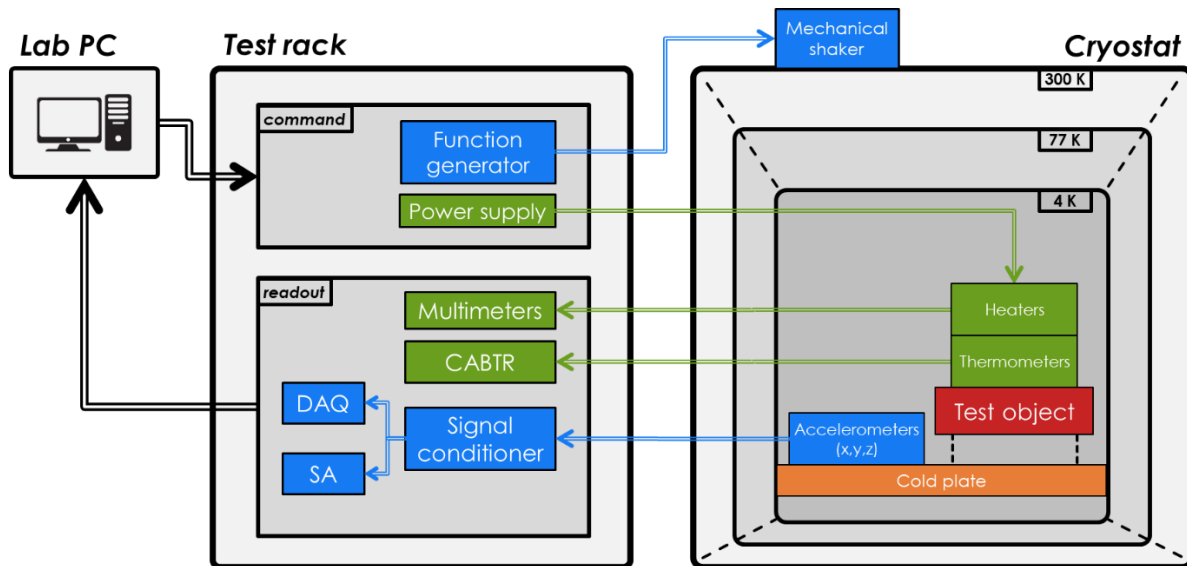


Figure 2.1 – Schematic of microvibration cryostat test bench (cold configuration)

The schematic shows the test bench in the cold configuration, where the cryostat is cooled and closed. Several acronyms are shown in the schematic. They are:

- CABTR: *Centrale d'acquisition Basses Températures Rapide*
- DAQ: *Data acquisition system*
- SA: *Spectrum analyser*

These readout systems will be covered in subsequent sections. The cryogenic environment (i.e. cryostat) is first described, before covering sensing and actuation, in both the mechanical and thermal domain.

2.2 Main test object

Before presenting the range of topics regarding measurement, it is necessary we present the main test object of this thesis, as we refer to it from here on throughout the dissertation.

The main test object is an elementary cryogenic device, a gas-gap heat switch. A small copper mass of ~10 grams can be thermally decoupled from a cold source by means of a thin-walled insulating titanium tube of ~5 cm. This tube still allows transmission of microvibrations to the insulated tip of the heat switch, making it an ideal device for studying microvibration dissipation. At our disposal are 2 identical units, henceforth named HS1 and HS2. For reasons we shall discuss, the majority of the tests presented in this thesis were carried out with HS2.

Heat switches are omnipresent in cryogenic systems. Gas-gap heat switches specifically, are ubiquitous in sub-Kelvin applications. Much like electrical switches, they are used to control the flow of energy, the difference being that this energy is a heat flux as opposed to an electrical current. The mechanical behaviour of the type studied in this thesis can be considered similar to a classical cantilever beam. In its simplest form, the heat switch is shown Figure 2.2. A cantilever beam is in fact a mass-spring type system often studied in mechanics, displaying mechanical resonances. External mechanical excitation in the form of vibrations may induce oscillations of the free end. In real systems these oscillations decay over time due to damping when excitation ceases, say from friction or material deformation, causing dissipation that degrades into heat. In the context of this thesis, these oscillations correspond to oscillations of the heat switch, due to induced microvibrations at its mechanical interface.

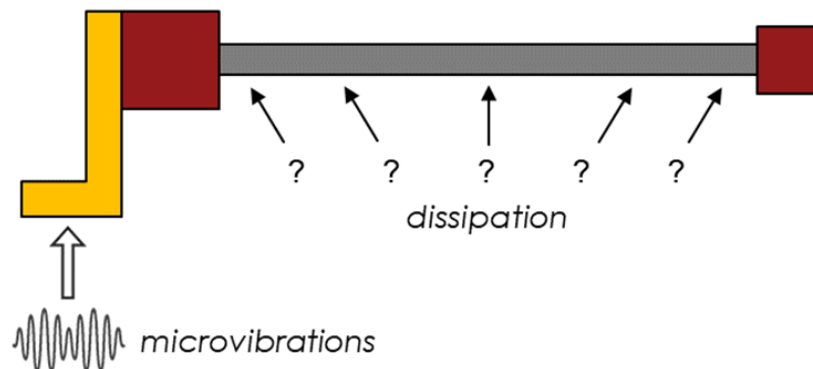


Figure 2.2 – Gas-gap heat switch microvibration dissipation

A heat switch was first chosen for this purpose as it was deemed simple enough to begin studying microvibration dissipation: microvibrations can be induced into an insulated mass (free end of the heat switch visible in the right part of Figure 2.2). In addition, the heat switch can by design be cooled fast yet be well insulated in cold conditions. For these reasons, we concluded that the heat switch would enable us to achieve the thesis objectives outlined at the end of chapter 1. The stabilisation and cooling times of the heat switch was also studied and is detailed in Appendix 1.

Its working principle and mechanical characteristics will now be presented. These are key aspects that highlight why it is suitable to study microvibration dissipation. The remainder of this section then discusses its mechanical resonance, measured and predicted analytically, and at various temperatures.

2.2.1 Working principle

A standard gas-gap heat switch²² is shown in Figure 2.3. Two copper components, the base and the tip of the heat switch, are thermally coupled or decoupled through the presence or absence of thermally conductive helium gas in a so-called gas gap. An active charcoal miniature sorption pump is controlled to desorb or adsorb said gas in the gas gap. The latter is a volume enclosed by a thin titanium tube that is fixed to the heat switch base at one end and the heat switch tip at the other. The tip and the base are otherwise not in contact. In the OFF mode, gas is adsorbed into the mini pump and the tip is insulated. In the ON mode, gas is desorbed and the tip is coupled to the base thanks to their large concentric surfaces. At 4.2 K, conductance between the tip and base, in the OFF mode and ON mode, is rated at 12 $\mu\text{W/K}$ and 40 mW/K respectively for our device. The ON/OFF ratio is thus roughly 3300.

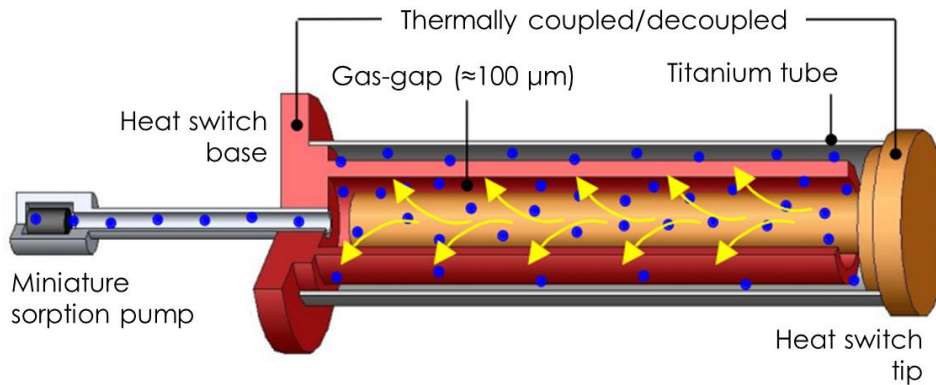


Figure 2.3 – Schematic of a standard gas-gap heat switch²²

Thus we have at our disposal a device that thermally decouples a copper mass (i.e. the heat switch tip in the right side of Figure 2.3), while still allowing mechanical transmission to induce microvibrations (both in the titanium tube and the copper tip). Furthermore, the heat switch tip can be cooled fast by switching to the ON mode (the cold source being the interface of the heat switch base). This makes a heat switch an ideal device for studying the dissipation due to induced microvibrations. For this thesis, 2 gas-gap heat switches are at our disposal. They shall henceforth be named HS1 and HS2. Given that we wish to induce microvibrations in them, it is also crucial to characterise them from a mechanical standpoint.

2.2.2 Mechanical characteristics

The key mechanical characteristic that we are interested in is the resonance frequency of the heat switch, specifically the 1st mode of the heat switch tip cantilever beam. Indeed, were dissipation due to induced microvibrations be present, we expect to observe it at resonance, where oscillations are greatest.

In the first test setup, the heat switches are held rigidly at their bases. The titanium tube extends from one side of the base and supports the heat switch tip. Extending to the opposite side is the miniature pump (not the core of this study). The side view of the exact heat switch model used is shown Figure 2.4. The tip extends 83 mm and the mini pump 45.5 mm from the rightmost surface of the base.

²² (Duband, 2009)

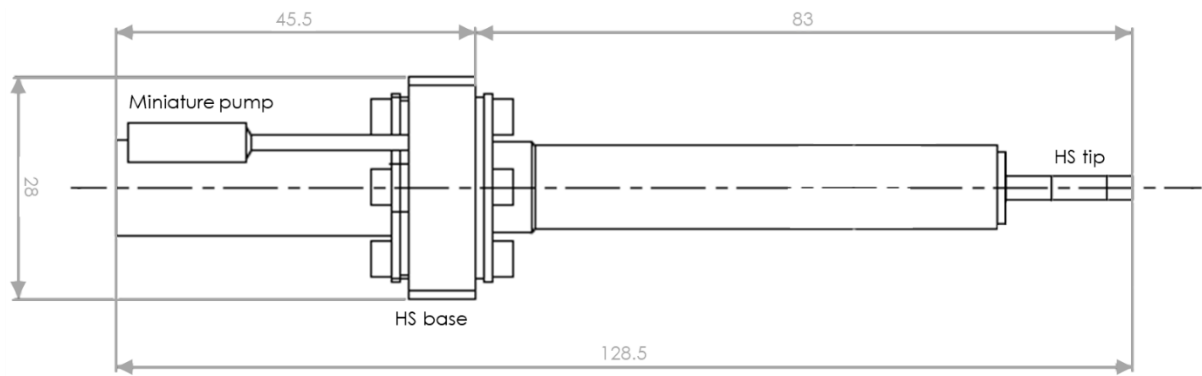


Figure 2.4 – Heat switch side view and key outer dimensions

Other mechanical parameters of the heat switch are summarised in Table 2.1. We use them next to estimate the cantilever beam resonance f_{res} , as well as for thermal calculations later in this chapter.

Table 2.1 – Mechanical parameters of heat switch

	Parameter	Value
<i>Titanium tube inner diameter</i>	d_e	7.2 mm
<i>Titanium tube outer diameter</i>	d_i	7.0 mm
<i>Titanium tube cross-section area</i>	$S = \pi \left(\frac{d_e^2}{4} - \frac{d_i^2}{4} \right)$	2.23 mm ²
<i>Titanium tube moment of inertia</i>	$I = \frac{\pi}{64} (d_e^4 - d_i^4)$	14.06 mm ⁴
<i>Titanium tube effective length</i>	L	50 mm
<i>Titanium Young modulus (at 300 K)</i>	E	110 GPa
<i>Mass HS base (copper)</i>	m_{base}	65.6 g
<i>Mass HS tip (copper)</i>	m_{tip}	11.9 g
<i>Mass instrumentation at tip (w/o steel)</i>	m_{instr}	4.5 g
<i>Mass instrumentation at tip (only steel)</i>	m_{steel}	2.6 g
<i>Mass total at tip</i>	$m = m_{tip} + m_{instr} + m_{steel}$	19.0 g

2.2.3 Resonance of a cantilever beam

Before measuring the natural frequencies of the heat switches, it is worth defining the 1st mode resonance of a cantilever beam analytically. Its definition varies slightly depending on the distribution of the load along its length. Regarding the heat switch, the mass of its titanium tube is 0.6 g, whereas the mass at its end is no less than 11.9 g (when no instrumentation is added). We can therefore reasonably assume to be in the case of a point load cantilever beam. The 1st mode resonance frequency is then given by:

$$f_{res} = \frac{1}{2\pi} \sqrt{\frac{3EI}{mL^3}} \quad (2.1)$$

Where E is the Young modulus, I the moment of inertia and L the length, of the element that provides mechanical stiffness. Parameter m is the mass of the point load. For the heat switch, mechanical stiffness is provided by the titanium tube and the point load is the total mass at the tip of the heat switch.

Indeed the resonances are later measured with the heat switch tips instrumented with thermometers and heaters. Naturally, this adds mass and so reduces the resonance frequency. As tip instrumentation is mandatory for dissipation measurement, calculation and measurement of the mechanical behaviour of the heat switch will be done in this configuration ($m = 19.0$ g).

Using the mechanical parameters of Table 2.1, the estimated 1st mode resonance frequency of the heat switch tip is 222.4 Hz, using a Young modulus of titanium at 300 K. As we shall now see, this slightly overestimates the measured resonance frequencies at room temperature.

2.2.4 Resonance of heat switch at room temperature

The 1st cantilever modes of HS1 and HS2 were measured using a laser sensor at ambient temperature (we detail all sensing and actuation equipment used in this thesis later in the chapter). With the flick of a finger we induce an impulse-like excitation, corresponding to white noise excitation. This inevitably excites the resonance that is identified from the FFT transform of the laser sensor output voltage. Figure 2.5 shows these tests for the heat switch tips and Figure 2.6 shows them miniature sorption pumps.

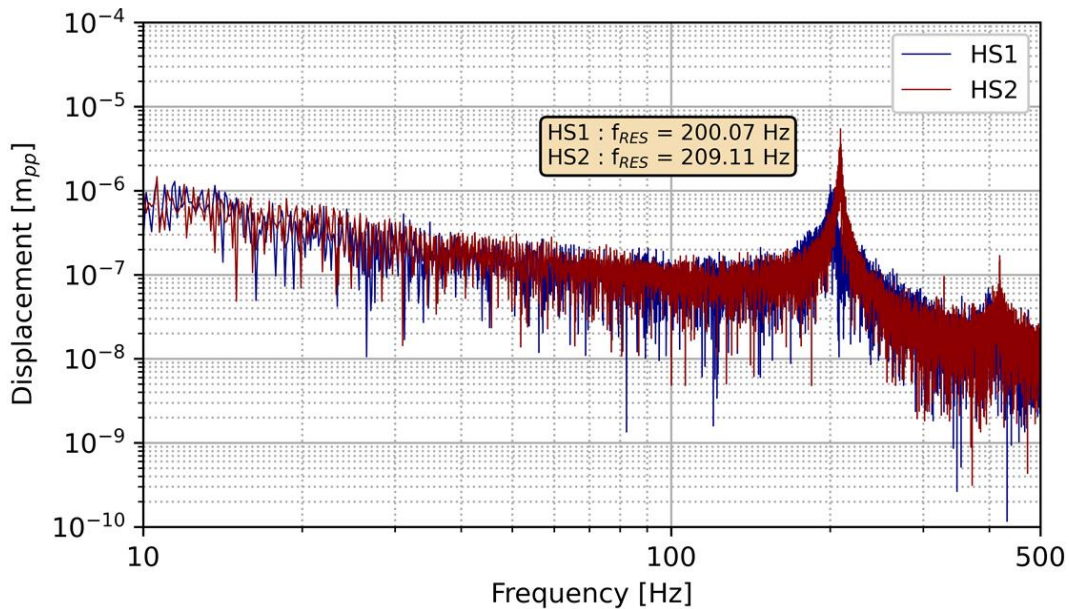


Figure 2.5 – 1st mode resonances of main HS tip cantilevers

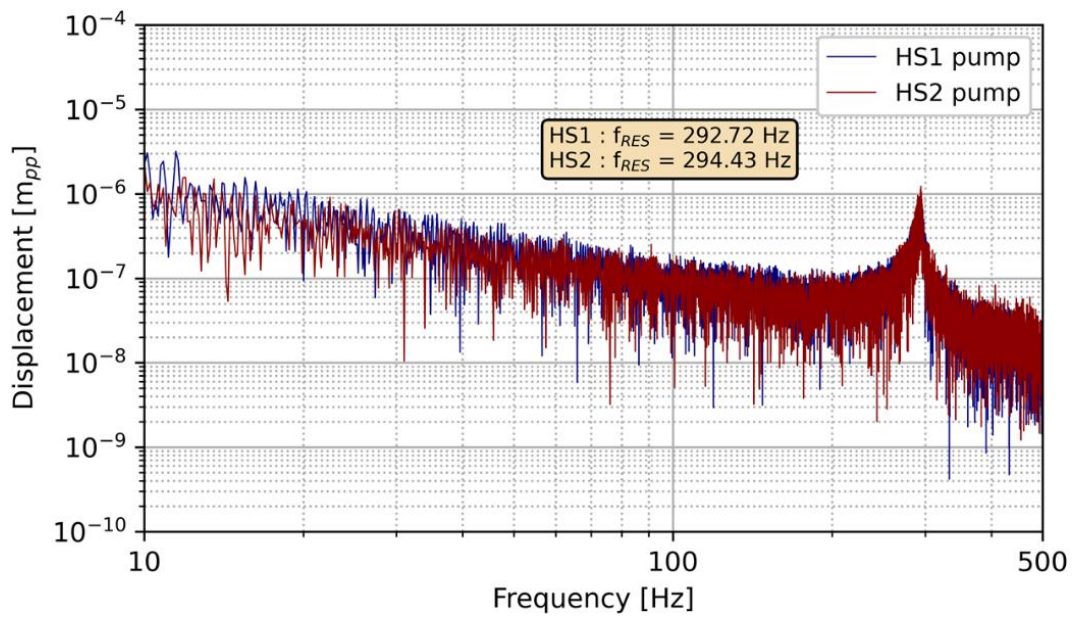


Figure 2.6 – 1st mode resonances of mini sorption pumps

The 1st mode natural frequencies found are given in Table 2.2, to within 1 Hz. The resonances of the mini pumps are sufficiently far from those of the heat switch tips that they do not interfere with each other.

Table 2.2 – Main heat switch resonances measured at ambient temperature

Resonance	HS1	HS2
<i>Heat switch tip</i>	200 Hz	209 Hz
<i>Mini sorption pump</i>	293 Hz	294 Hz

We note that the resonance of the tips was analytically overestimated by about 22 Hz for HS1 and 13 Hz for HS2. This could be explained by having underestimated the total mass or effective length of the titanium tube, or by having overestimated the Young modulus or moment of inertia. As per equation (2.1) however, the greatest impact on the resonance frequency is the effective length L , a parameter that is difficult to determine in practice due to the assembly process (brazing). We note that using 51 mm instead of 50 mm as a value for L reduces the estimated resonance frequency to 215.9 Hz (a shift of 6.5 Hz). The experimental observations are nevertheless in good agreement with the analytical estimation of the resonance frequency at ambient temperature. We also note that the dispersion in measured frequencies between HS1 and HS2 is similar to the difference between the model and measured frequency of HS2.

This resonance was found to increase when the heat switch was cooled. This is explained by an increase in the Young modulus of titanium, as it is the titanium tube that defines the mechanical stiffness of the heat switch. Thermal contraction of the titanium tube may also increase the resonance. In our estimation of its resonance, we used the Young modulus of titanium at 300 K. To estimate the resonance at say 1.2 K, we must therefore discuss the evolution of this Young modulus as we cool our assembly. We will then be able to estimate the heat switch resonance in cryogenic conditions.

2.2.5 Resonance of heat switch at 1.2 K

The Young modulus for 2 titanium alloys is found to decrease with increasing temperature (as expected for metallic materials), showing flatness below ~20 K and approaching linear behaviour at higher temperatures²³, as shown in Figure 2.7.

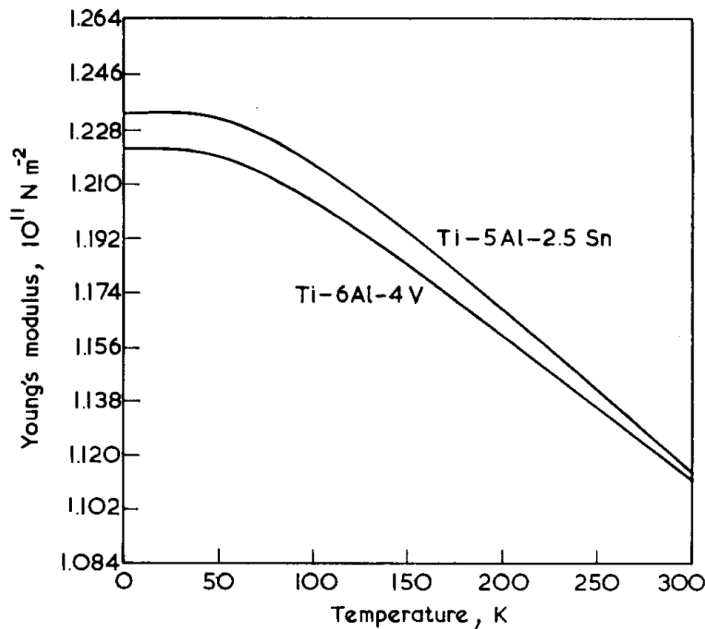


Figure 2.7 – Evolution of Young modulus with temperature of 2 titanium alloys²³

We note in Table 2.3 the change in the Young modulus of the titanium alloy used, namely Ti-6Al-4V (or TA6V), across 3 temperature ranges. Given that the Young modulus is proportional to mechanical stiffness and that resonance frequency is proportional to the square root of this stiffness as per equation (2.1), we also deduce the change in the resonance frequency. It is largest from 300 K to 100 K. Decreasing the temperature further does not drastically increase resonance frequency.

Table 2.3 – Young modulus percentage change with temperature for titanium (TA6V)

<i>Temperature change</i>	<i>Young modulus increase</i>	<i>Resonance frequency increase</i>
300 K to 100 K	~11.6%	~5.6%
100 K to 4.2 K	~1.1%	~0.5%
4.2 K to 1.2 K	~0%	~0%

As we shall see throughout this chapter, the resonance frequency increase of our heat switch when cooled falls within the changes predicted here, that is to say within ~6% of the resonance at 300 K.

From studies performed by NASA & Martin Marietta Materials on the properties of cryogenically cooled materials²⁴, similar figures can be given for stainless steel. Alloys of stainless steel are used for the

²³ (Naimon, Weston, & Ledbetter, 1974)

²⁴ (Schwartzberg & Kiefer, 1975)

structural supports and cryogen reservoirs of the cryostats used in this thesis. Deducing the change in its Young modulus helps gauge its contribution to observed changes in cryostat resonance frequencies due to temperature changes. The findings for alloy 17-7PH (or 1.4568 as per the European designation) are given in Table 2.4. Our cryostat in fact mostly contains another common steel grade (304 stainless steel) that is not shown here but whose characteristics are similar. The figures presented nevertheless provide an indication as to the susceptibility of the mechanical behaviour of our cryostat to temperature changes.

Table 2.4 – Young modulus percentage change with temperature for stainless steel alloy (1.4568)

<i>Temperature change</i>	<i>Young modulus increase</i>
293 K to 77 K	~12.0% (184 to 206 GPa)
293 K to 20 K	~21.7% (184 to 224 GPa)

2.3 Other test object

Another test object is worth presenting here, which at the time of writing is already designed and built, but has not yet been tested with regards to microvibrations dissipation.

The test object is shown in Figure 2.8. It is a Kevlar string suspension system that serves as a mockup of an ADR cooling stage. Its mobile mass is an aluminium rod held at its 2 ends that is roughly 20 cm in length and 3 cm in diameter. Its larger mass (315 g) compared to the heat switch (19 g) allows it to be more heavily instrumented, for example with several cryogenic accelerometers as shown, to measure the motion of the mobile mass when cooled and subjected to microvibrations, in multiple degrees of freedom.

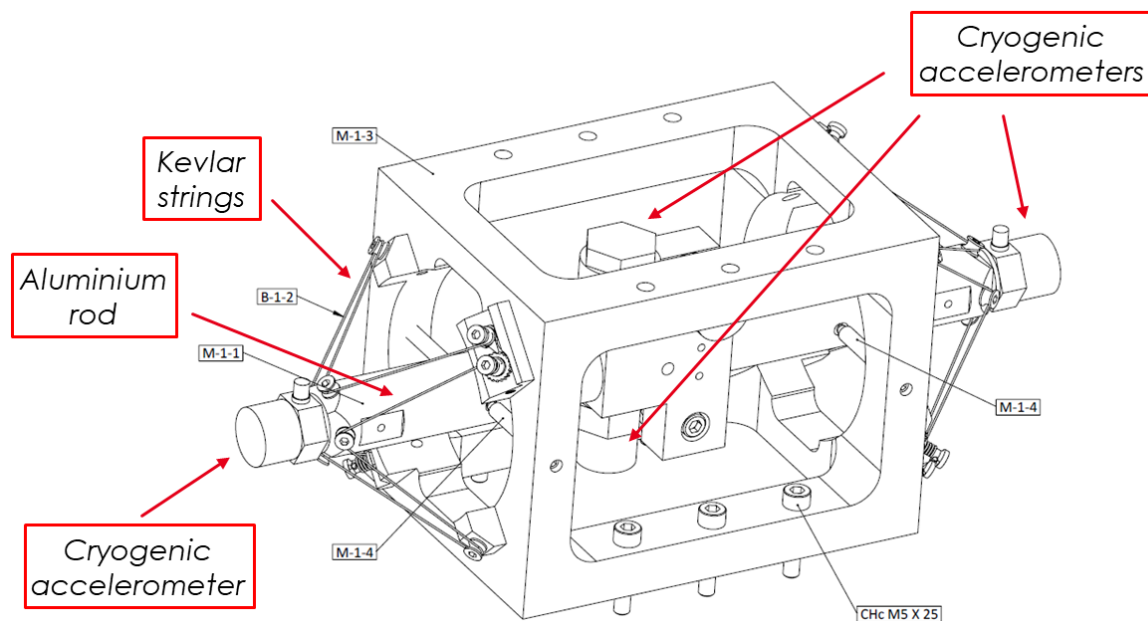


Figure 2.8 – Technical drawing of new test object based on a Kevlar string suspension system

The microvibration dissipation behaviour of this Kevlar suspension system is expected to differ compared to the heat switch, for example through stick-slip motion but also due to the very different nature of Kevlar

compared to metals. At the time of writing, this new test object had not yet been cooled and tested for microvibration to observe dissipation. It is therefore not discussed further in this dissertation.

2.4 Physical quantities

It is now useful to restate the physical quantities studied with our main test object, the gas-gap heat switch, having already been introduced as parameters in the models of chapter 1. These key quantities are:

- Acceleration (injected at test object interface): a [g]
- Power dissipation (due to induced microvibrations): P [W]
- Temperature (of test object or component): T [K]
- Displacement (of test object or component): x [m]

As mentioned in the previous chapter, microvibrations refer to small-scale mechanical vibrations. Bearing in mind that range definitions vary slightly depending on applications, typical microvibration ranges are:

- Typical acceleration range: $10 \mu g$ to $10 mg$
- Typical frequency range: $1 Hz$ to $1 kHz$

For our work and given the capabilities of our test bench and detection chain, we will work with slightly different ranges. These will be presented and justified in this chapter.

2.5 Cryogenic environment

For the study of microvibration dissipation in a cryogenic environment, we wish to mitigate to the fullest uncontrolled microvibration sources. A helium bath cryostat, with no moving parts (besides fluid motion and boiling) as opposed to a cryostat cooled by mechanical cryocoolers, is an ideal candidate. As discussed later, *controlled* microvibrations are nevertheless induced to quantify the effects of acceleration on the power dissipation due to microvibrations and the temperature of a test object. For this, we must properly account for the environment when setting up the test bench, as well as for the interpretation of experiments made with it.

2.5.1 Helium bath cryostat

Figure 2.9 shows one of two helium bath cryostats used throughout the thesis. Key characteristics of both will be summarised later. The cryostats can be placed in a so-called upside down (left) or upright (right) position, by means of a crank mechanism. This allows easy-of-access to the test object when the cryostat is open, as will be detailed further in this section.

The cryostat shown and from now on detailed is the larger of the two but both have similar thermal and mechanical architectures. Having served to test sub-Kelvin flight models for the Herschel telescope instruments prior to its launch in 2009, the cryostat is henceforth referred to as the “Herschel cryostat”.



Figure 2.9 – Herschel cryostat in the upside down position closed (left) and upright position open (right)

The Herschel cryostat’s main structure is held by 2 collinear pivots held by a support structure on wheels. The cryostat can be pivoted to switch between the upside down (when closed and cold) and upright positions (when open) as shown in Figure 2.9. The cryostat can also be tilted at an intermediate angle if needed, to for example study gravity direction effects on a test object (not of interest in this thesis).

It is worth highlighting at this point the definition of the principal axes used throughout this thesis. The vertical direction is always the z direction and the cryostat rotation axis the x direction. This convention is used throughout this thesis, for both cryostats.

To present the functioning of the cryostat, we begin with the outermost part: the so-called vacuum vessel and support structure. The cryostat is sealed at this level when closed, allowing a vacuum to be pumped inside. This fully reduces convection effects and drastically reduces conduction effects in the gas, increasing insulation. The outermost part of the vacuum vessel is always at ambient temperature around 300 K (see schematic of Figure 2.1). Inside we find the so-called 77 K thermal screen, thermalized to a liquid nitrogen reservoir. This thermal screen encapsulates the 4.2 K environment, shielding it from 300 K radiation. The so-called 4.2 K thermal screen is thermalized to a liquid helium-4 reservoir and surrounds the test environment, shielding it from 77 K radiation. Successive thermal screens limit radiation to the test environment that is encapsulated by the 4.2 K thermal screen. The 77 K screen limits the helium boil-off rate. According to the Stefan-Boltzmann law, the radiation per unit surface area P of a body of emissivity ϵ and temperature T , and Stefan-Boltzmann constant σ , is given by equation (2.2).

$$P = \varepsilon \sigma T^4 \quad (2.2)$$

Were the 4.2 K thermal screen be subjected to 300 K radiation rather than 77 K radiation, the corresponding power absorbed by it would be over 2 orders of magnitude greater: $300^4/77^4 \approx 230$. This highlights the importance of successive thermal screens and cooling stages in the thermal architecture of the cryostat (and cryogenic systems in general).

Both the nitrogen reservoir (with a capacity of 60 L) and helium reservoir (with a capacity of 32 L) are fillable from the exterior through dedicated inlet pipelines. Each reservoir also has outlet pipelines for the respective nitrogen and helium vapours due to bath boiling. These are labelled, together with the thermal screens and support structure discussed on the schematic shown in Figure 2.10. The liquid nitrogen and liquid helium inlets are labelled LN2 and LHe respectively, whereas the gaseous nitrogen and gaseous helium outlets are labelled GN2 and GHe respectively.

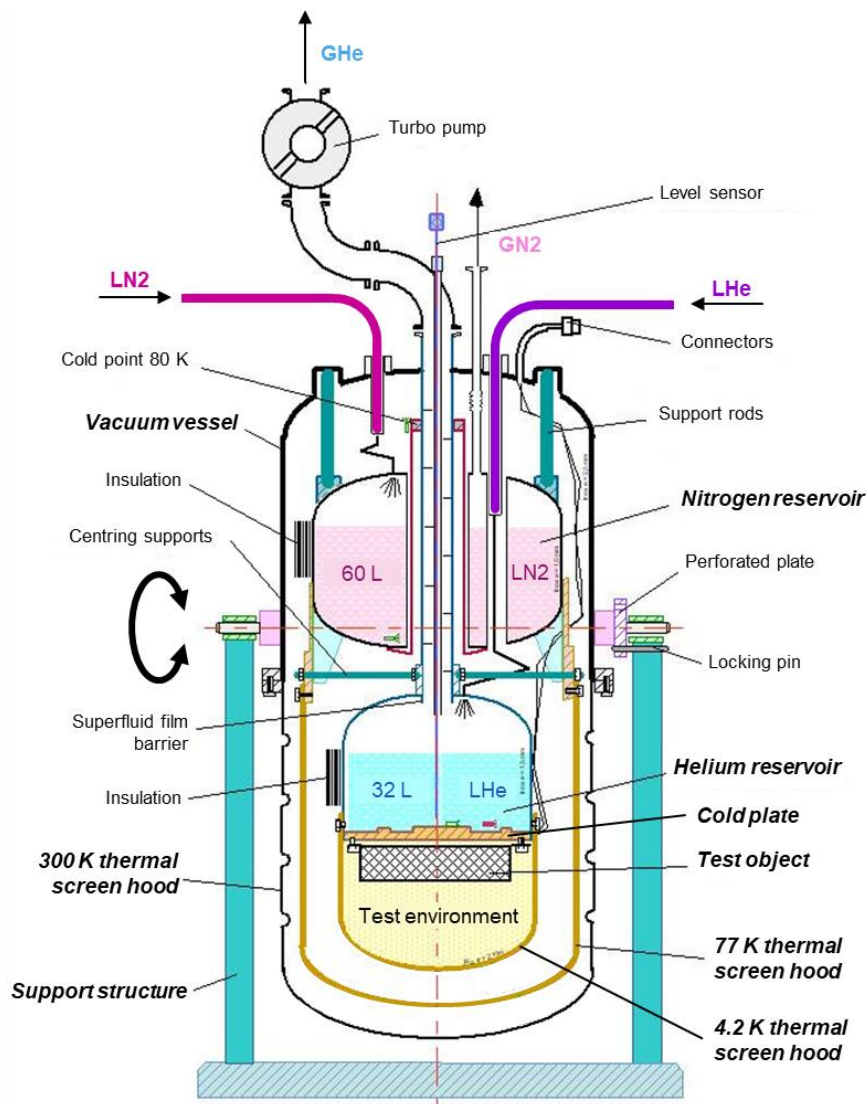


Figure 2.10 – Schematic of Herschel cryostat in upside down position

On Figure 2.10 are labelled the support structure, vacuum vessel, thermal screens and cryogen reservoirs. The helium reservoir is closed by a gold-plated circular copper plate, the so-called cold plate.

The test environment is the cylindrical volume that sits, in the cryostat upright position, directly atop this cold plate. This is shown in more detail in Figure 2.11. The test object (visible as 2 gas-gap heat switches) is fixed and thermalized to the cold plate. The thermal screens without their hoods are also labelled here.

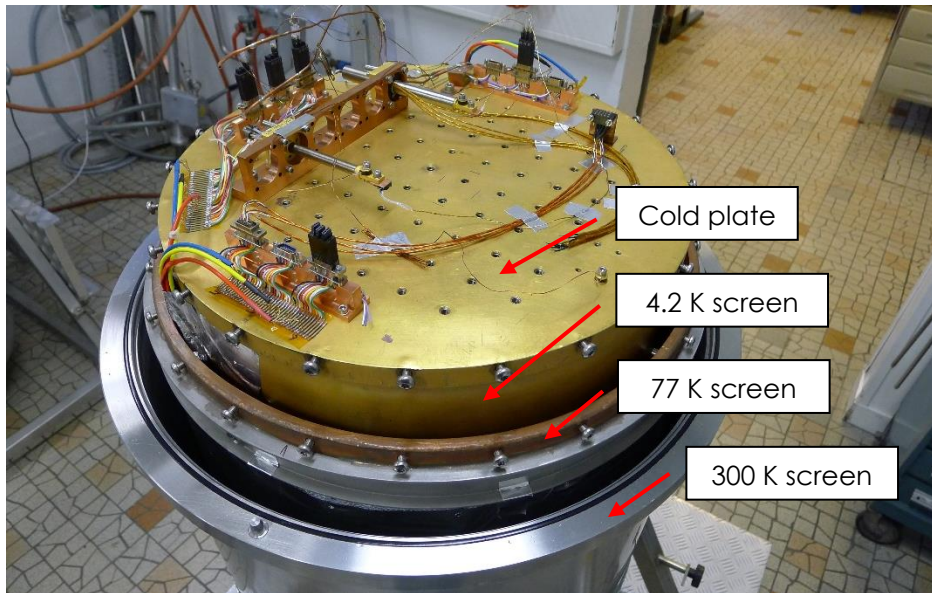


Figure 2.11 – Herschel cryostat open and thermal screen hoods removed

The 300 K screen and 77 K thermal screen are always at ambient temperature and 77 K respectively, the latter determined by the temperature of LN2 at ambient pressure. The temperature of the helium reservoir – and so 4.2 K thermal screen and cold plate – is *at most* 4.2 K. As we shall see later, the cryostats have two modes of operation allowing the cold plate temperature to drop below 4.2 K.

The second so called “Tedi cryostat” is the smaller of the two used during this thesis. Its architecture and functioning is almost identical to that of the Herschel cryostat described thus far. It is worth highlighting though, some key differences between the two cryostats. These are summarised in Table 2.5.

Table 2.5 – Key characteristics of Herschel cryostat and Tedi cryostats

Name	<i>Herschel cryostat</i>	<i>Tedi cryostat</i>
<i>Size (W x H)</i>	Ø45 cm x 180 cm	Ø45 cm x 145 cm
<i>Cold plate diameter</i>	Ø38 cm	Ø30 cm
<i>Nitrogen reservoir</i>	60 L	30 L
<i>Helium reservoir</i>	32 L	16 L
<i>Cold autonomy (at 4.2 K)</i>	~8 days	~2 days

Now we will cover the steps involved in the cooling procedure of the cryostat.

2.5.2 Cooling procedure

The main steps to cool the cryostat are: (i) pumping the vacuum vessel, (ii) cooling to 77 K and (iii) cooling to 4.2 K. A 4th step may also follow for cooler temperatures still, by pumping the helium bath.

First, a primary pump is used to reduce the internal pressure of the vacuum vessel from 1 bar to 1 mbar. With the help of a secondary pump (generally a turbo-molecular pump) first at lower speed, the pressure is reduced to 10^{-3} mbar. The secondary pump is then set to full speed ($\sim 24\,000$ RPM), further reducing the internal pressure to as low as 10^{-5} mbar. In this way, convection effects are fully eliminated. At this stage, thermal insulation is thus greatly improved and cooling can begin.

Cooling to 77 K requires the use of liquid nitrogen (LN₂). The cryostat's nitrogen reservoir is filled by means of an external dewar of LN₂. A limited amount of LN₂ may also be added to the helium reservoir to precool it before the addition of liquid helium (LHe). This step avoids unnecessarily consuming helium before reaching 77 K, a wise choice given its significantly higher cost and lower latent heat compared to nitrogen. Adding cryogen to the helium reservoir already at this stage also cools the test environment faster, as the cold plate is thermalized to the helium reservoir. The nitrogen reservoir is filled and the helium reservoir is cooled to 77 K. The latter must be empty before the next phase of cooling to ensure that the entirety of the LN₂ has boiled off and evacuated as gaseous nitrogen. When this point is reached and the test environment is at 77 K, cooling to 4.2 K can begin.

Cooling to 4.2 K requires liquid helium (LHe). The helium reservoir is filled by an external dewar too, now of liquid helium-4 (that boils at 4.2 K at ambient pressure). At first, when subjected to a precooled helium reservoir at 77 K, the LHe evaporates. Gaseous cooled helium cools said reservoir until a temperature of 4.2 K is reached. At this temperature the first drops of LHe appear and the helium reservoir fills with LHe. Cooling is complete once the helium reservoir is full and the test object and its environment reach 4.2 K.

At this stage, the primary and secondary pumps may also be switched off. This is particularly necessary regarding the study of microvibrations, where a “mechanically clean” environment is needed. The vacuum is in part maintained by cryo-pumping or cryo-adsorption, whereby atoms are condensed and trapped on the cold surfaces. Despite achieving a high quality vacuum, heat flux losses still exist between the various cold stages of the cryostat and the ambient environment, through solid conduction and radiation. As such, the cryogenes will slowly consume themselves, limiting the available experimentation time at 4.2 K. As summarised in Table 2.5, the larger Herschel cryostat consumes its cryogenes at 4.2 K in about 8 days and the smaller Tedi cryostat in roughly 2 days, before a refilling of the reservoirs is necessary.

Having now covered cooling to 4.2 K, the following subsection will detail how we can cool the helium bath further still, by pumping it. The motivation for doing so is also discussed.

2.5.3 Pumped bath

The microvibration cryostat test bench shown Figure 2.1 and discussed until now describes the steps to cool the cold plate and test environment to 4.2 K. In this section we shall see how they can be cooled further, by pumping the helium bath, as well as cover the motivation for working at lower temperature.

Two stable isotopes of helium exist: helium-4 and helium-3. The most common and widely used isotope is helium-4, as the cryogen of choice in helium-bath cryostats. Helium-3 nevertheless does have other cryogenic applications, including usage in dilution refrigerators and sorption coolers, though these are beyond the scope of this thesis. Figure 2.12 shows the phase diagram for the more common helium-4 isotope, down to absolute zero and a pressure of 1 Pa = 10^{-2} mbar.

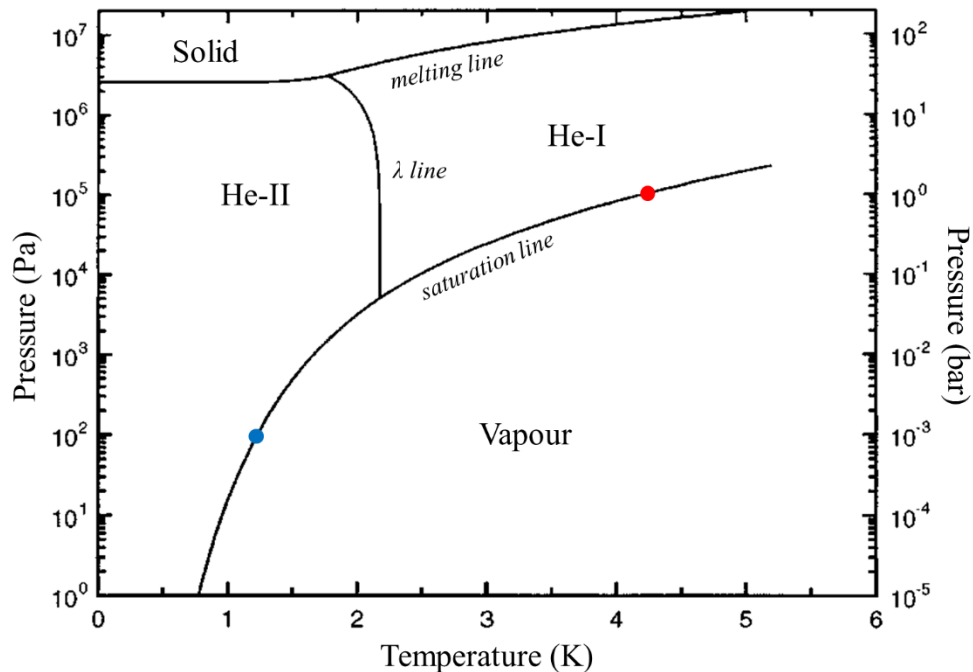


Figure 2.12 – Phase diagram of helium-4 ²⁵

The vapour phase is visible at low pressure and higher temperature (bottom right region), the solid phase at high pressure and low temperature (top left region), and the liquid phases (He I and He II) in between. The solid and liquid phases are separated by the melting line. The vapour and liquid phases are separated by the saturation line, where helium is in a state of equilibrium between the vapour state and liquid state. This equilibrium is temperature and pressure dependant. As with water that boils at 100°C at 1 atm and boils at a lower temperature at lower pressure, so too does helium-4.

The liquid phase is further divided by the so-called lambda line, separating normal liquid helium (He-I) and superfluid helium (He-II). The properties of helium between these 2 states differs considerably. For He-II its viscosity is near-zero, allowing it to flow almost frictionless along a surface and to permeate very small spaces (e.g. windings of superconducting magnets).

The cooling power available to the cold plate in the cryostat is drawn from the latent heat of liquid helium in a saturated state. Increased heat input (e.g. from the accelerometers due to joule heating) will cause the helium bath to boil off faster, though remain at a constant temperature. This temperature is determined by the pressure above the bath. By slowly varying this pressure, we move along the saturation line and in doing so vary the saturated helium (and so bath) temperature. The minimum achievable temperature is governed by the quality of the partial vacuum achievable by the pumps at our disposal.

²⁵ (J. Donnelly & F. Barenghi, 1998)

At ambient pressure at ~ 1 bar, the saturated helium temperature is close to 4.2 K (marked with a red dot in Figure 2.12). In our cryostat, when the helium bath is pumped, we generally attain a vacuum pressure below 1 mbar, corresponding to a bath temperature close to 1.2 K (marked with blue dot in Figure 2.12). Thus, in this way we have at our disposal a cryostat that provides a 1.2 K interface at the cold plate.

The main advantage of the pumped bath configuration is improved temperature stability, ensured through improved bath pressure stability. In the ambient pressure configuration the helium vapour is recovered in storage tanks before being reliquefied on site. In the lab where the experiments described in this thesis were carried out, a single storage tank collects the helium vapour from several sources at different locations simultaneously. This network of piping couples the helium bath pressure of the Herschel cryostat to several other sources. The bath pressure is thus unstable and therefore so too the saturated helium temperature. When the bath is pumped, improved pressure stability is ensured by the vacuum pump to which it is now coupled, having been decoupled from the other perturbing helium vapour sources.

The impact of pumping the helium bath on pressure and so temperature stability will be covered in the following chapter, when discussing the characteristics and limitations of the cryogenic environment. Having now presented the cryogenic environment (i.e. the cryostat), its cooling procedure and the motivation for pumping the helium bath, we shall now cover the other broad topics related to measurement. We begin with sensing and actuation, segmented into mechanical and thermal types.

2.6 Mechanical sensing and actuation

For the study of the dissipation of microvibrations, sensing and actuation is needed in and around the test object under study placed in the cryogenic environment. Regarding mechanical sensing and actuation used to study microvibration dissipation at low temperatures, this section is divided into several topics: *voltage, acceleration, displacement, full measurement chain noise and mechanical excitation*.

2.6.1 Voltage

As we shall see, the mechanical sensors used output voltage signals. The quality of voltage measurement is essential if we wish to achieve accurate and precise acceleration and displacement measurements. The voltage measurement systems used for this thesis are a data acquisition system (DAQ) and a spectrum analyser. These will now each be presented, followed by a discussion on the observed impacts of sampling frequency and test duration on voltage measurement.

2.6.1.1 Data acquisition system

Multi-channel and high sample rate voltage measurement is performed using a DAQ system. The model used is the NI 4431 shown in Figure 2.13. It has 4 analog inputs and 1 analog output. Integrated analog-to-digital conversion electronics allow analog signals from the cryogenic accelerometers (having passed through the signal conditioner) and displacement sensor to be digitised and recorded on the lab PC, as depicted in Figure 2.1. Some key specifications are shown in Table 2.6.

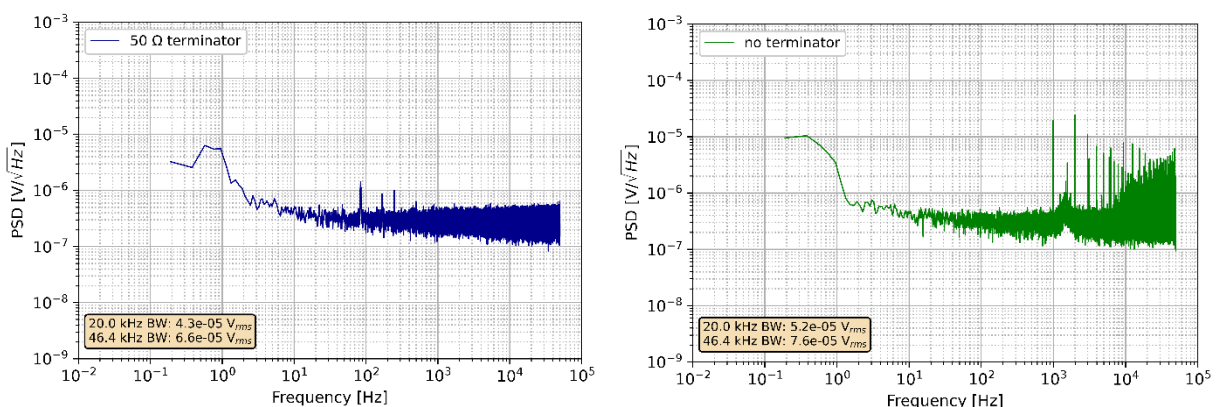


Figure 2.13 – DAQ system (NI 4431)

Table 2.6 – Key functional specifications of DAQ system

Name	Value
Sample rate range	1 kS/s to 102.4 kS/s
Input range	$\pm 10 V_{pk}$
ADC resolution	24 bits
AC coupling cutoff (-3 dB)	0.8 Hz
AC coupling cutoff (-0.1 dB)	6 Hz
Broadband noise (0-20 kHz)	55 μV_{rms}
Broadband noise (0-46.4 kHz)	75 μV_{rms}

Noise level specifications can be compared to measured levels. For broadband noise measurement, the highest sample rate is set (i.e. 102.4 kS/s) and a test duration T of 21 s. The measurements are post-processed with Welch²⁶ segment lengths of $T/4$ and 50% overlap. The minimum observable frequency then becomes $4/T = 190$ mHz. The resulting noise power spectral densities (PSD) are shown with and without the use of a 50 Ω terminator in Figure 2.14.

Figure 2.14 – DAQ noise power spectral densities with 50 Ω terminator (left) and without 50 Ω terminator (right)

²⁶ Welch's method reduces noise in estimated power spectral densities, in exchange for reducing frequency resolution. It performs Fourier transforms on successive segments of length $L < T$ that overlap from 0% to 100%.

Using a terminator at the analog BNC input of the DAQ system limits the detection of parasitic charges from the environment that may influence the measurement. We note from Figure 2.14 the effect on the broadband noise levels of the use of such a terminator, summarised in Table 2.7. The measured spectral noise level over our band of interest (100-1000 Hz) is also shown.

We observe the influence of environmental noise of the order of $10 \mu\text{V}_{\text{rms}}$ (noise level with terminator versus without terminator), representing roughly 15-20% of the DAQ system's broadband electrical noise. We also observe measured broadband noise levels below (with terminator) or in line with (no terminator) the DAQ system specifications listed in Table 2.6.

Table 2.7 – Measured noise levels of DAQ system

Name	w/ terminator	w/o terminator
Broadband noise (0-20 kHz)	$43 \mu\text{V}_{\text{rms}}$	$52 \mu\text{V}_{\text{rms}}$
Broadband noise (0-46.4 kHz)	$66 \mu\text{V}_{\text{rms}}$	$76 \mu\text{V}_{\text{rms}}$
Spectral noise (100-1000 Hz)	$0.5 \mu\text{V}/\sqrt{\text{Hz}}$	

This noise level represents the lowest broadband voltage the DAQ system is capable of measuring. Assuming a noiseless accelerometer having a sensitivity of 100 mV/g , a broadband noise level of $43 \mu\text{V}_{\text{rms}}$ (for 0-20 kHz) corresponds to an acceleration noise floor of $430 \mu\text{g}_{\text{rms}}$. Reducing the bandwidth can lower this noise floor considerably, as we shall see.

2.6.1.2 Spectrum analyser

Whilst the DAQ system is used to exploit its multi-channel synchronisation capabilities and high sample rates, it is not used for real-time monitoring. To monitor the spectral content of a signal in quasi real-time, say from a cryogenic accelerometer, a spectrum analyser is used.

Mechanical measurements would be carried out “blindly” without the spectrum analyser, making the experimental procedures longer and less flexible. The spectrum analyser used is a Stanford Research Systems SR770, shown in Figure 2.15.



Figure 2.15 – Spectrum analyser (Stanford Research Systems SR770)

Contrary to the DAQ system, the spectrum analyser only has 1 single-ended analog input and 1 differential analog input, for quasi real-time monitoring. Several key specifications are shown in Table 2.8.

Table 2.8 – Key functional specifications of spectrum analyser

Name	Value
<i>Bandwidth</i>	DC to 100 kHz
<i>Input range (full-scale)</i>	0.1 mV _p to 50 V _p
<i>Spectral span (min/max)</i>	191 mHz / 100 kHz
<i>Spectral resolution (max/min)</i>	478 μ Hz / 250 Hz
<i>Spectral noise (1000 Hz)</i>	5 nV _{rms} / $\sqrt{\text{Hz}}$

We note that the measurement bandwidth of the spectrum analyser is about double that of the DAQ system, at 100 kHz and roughly 50 kHz (having a maximum sample frequency of 102.4 kS/s) respectively. At its highest resolution, the spectrum analyser measures frequencies to within ~ 0.5 mHz (its lowest span of 191 mHz divided into 400 spectral bins) and a very low noise floor of 5 nV_{rms}/ $\sqrt{\text{Hz}}$. These points make it a good system for studying low level vibrations and sharp (i.e. lightly damped) resonances precisely.

Before covering more upstream components of the voltage measurement chain, namely acceleration and displacement sensors, we shall cover some impacts observed of sampling frequency and test duration on measured power spectral densities.

2.6.1.3 Impact of sampling frequency

The DAQ system is capable of sampling rates from 1 kS/s to 102.4 kS/s. To observe the impact of sampling frequency, a noise test was sampled at 100 kS/s and down sampled to varying degrees. In this way lower sample rate signals are approximated from a single measurement, ensuring any observed differences are only due to varying sampling rates. The original 100 kS/s measurement down sampled by a factor 2 yields a 50 kS/s signal, by a factor 4 a 25 kS/s signal and so on. The noise spectral densities of these tests, having applied Welch segment lengths of T/2 and 50% overlap, are shown in Figure 2.16.

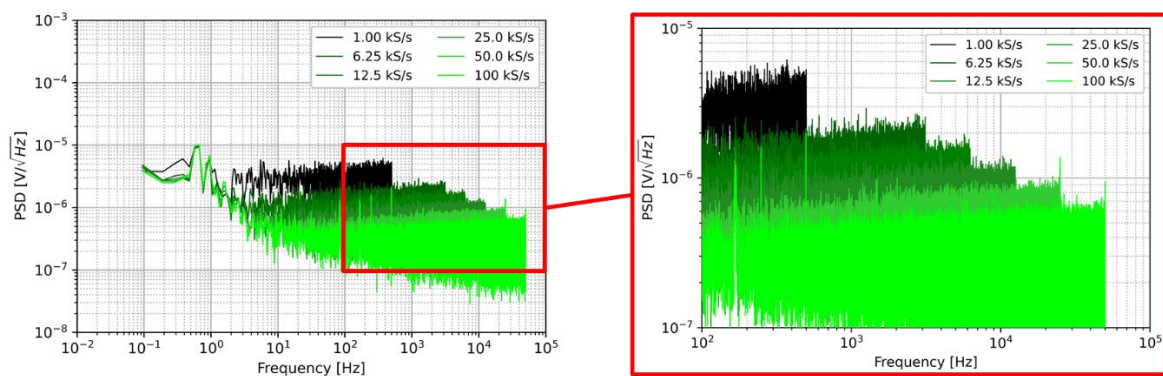


Figure 2.16 – DAQ noise measurement for various sample rates

The noise floor was found to fall for increasing sample rates, by roughly half an order of magnitude in the 100-1000 Hz band between sample rates 1 kS/s and 100 kS/s (likely due to aliasing). Only for broadband noise measurements is a higher sample rate necessary. For all other tests carried out in this thesis, we accept a slightly higher noise floor and choose a lower sample rate, given our typical working range of 10 Hz to 500 Hz. Sampling say at 1 kS/s naturally filters out noise above the Nyquist frequency of 500

Hz. A sample rate well below 100 kS/s also facilitates data manipulation, particularly for long duration tests when we wish to observe low frequency signals (i.e. drift) or study acceleration stability over time.

2.6.1.4 Impact of test duration

For a test duration of T seconds and applying Welch segment lengths of $T/2$ and 50% overlap, the minimum observable frequency is $2/T$ Hz. This impact is shown in Figure 2.17, where a noise measurement at 1 kS/s is shown, as well as the same time signal truncated to one tenth and one hundredth of its original length. We note that for a shorter signal duration, the minimum frequency is higher. When calculating the full bandwidth noise (i.e. 0-500 Hz), we also note almost identical broadband noise levels: 45.1 μ V (full duration), 45.0 μ V ($1/10$ duration) and 45.0 μ V ($1/100$ duration). This only corresponds to a 0.2% decrease.

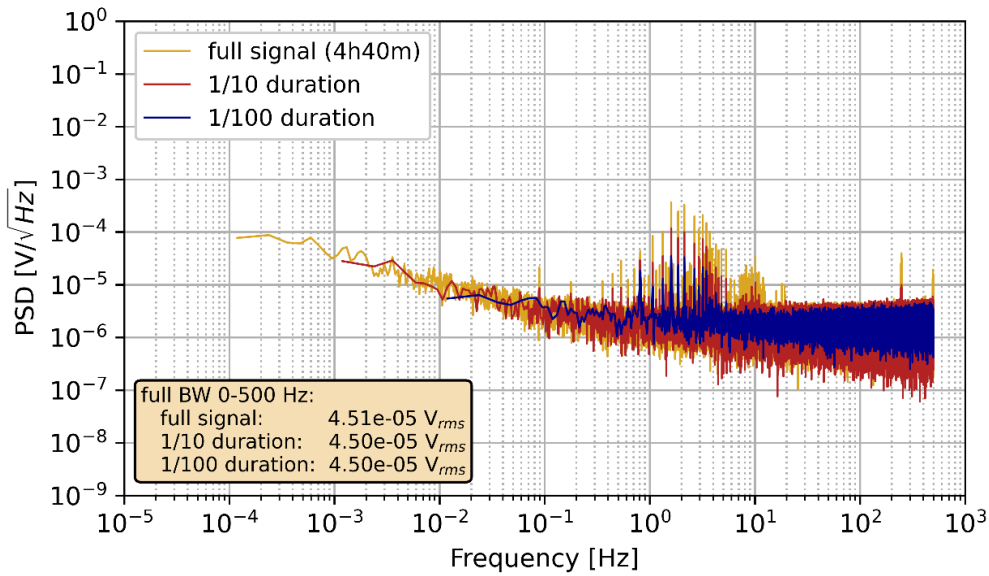


Figure 2.17 – DAQ noise spectral density for various test durations

Given that the lower bound of the typical frequency range of microvibrations is 10 Hz, the test duration is rarely a limiting test parameter. For fixed frequency excitation, amplitude resolution increases with the number of periods measured. Figure 2.18 shows a 10 mV_{pk} signal at 100 mHz sampled for 1, 10 and 100 periods. We note a high amplitude resolution already at 100 periods. We also note a gradual convergence to 10 mV_{pk} of the deduced V_{rms} values by integration of the PSDs: from 7.31 mV_{rms} (~ 10.34 mV_{pk}) for 1 period measured to 7.12 mV_{rms} (~ 10.07 mV_{pk}) for 100 periods measured. The sufficient number of periods measured for good amplitude resolution is also dependant on the signal-to-noise ratio (SNR) however.

The SNR happens to be very high in this example, given the signal is generated by a function generator. Regarding the real mechanical systems studied in this thesis that have lower SNRs, we find that 1000 periods of measurement provides satisfactory amplitude resolution. Hence as a ballpark figure we note that a minimum test duration of 100 s is needed (i.e. 1000 periods of a 10 Hz signal), for good voltage (i.e. acceleration) resolution.

However, for measurements involving frequency sweeps, amplitude resolution and spectral resolution also worsen as sweep speed increases. The highest acceptable sweep speed is ultimately dictated by thermal constraints (i.e. high thermal time constants) rather than mechanical constraints or data

acquisition factors such as those discussed above. The choice and effect of sweep speeds on thermal measurement are discussed in the chapter 4, discussing the measurement methods.

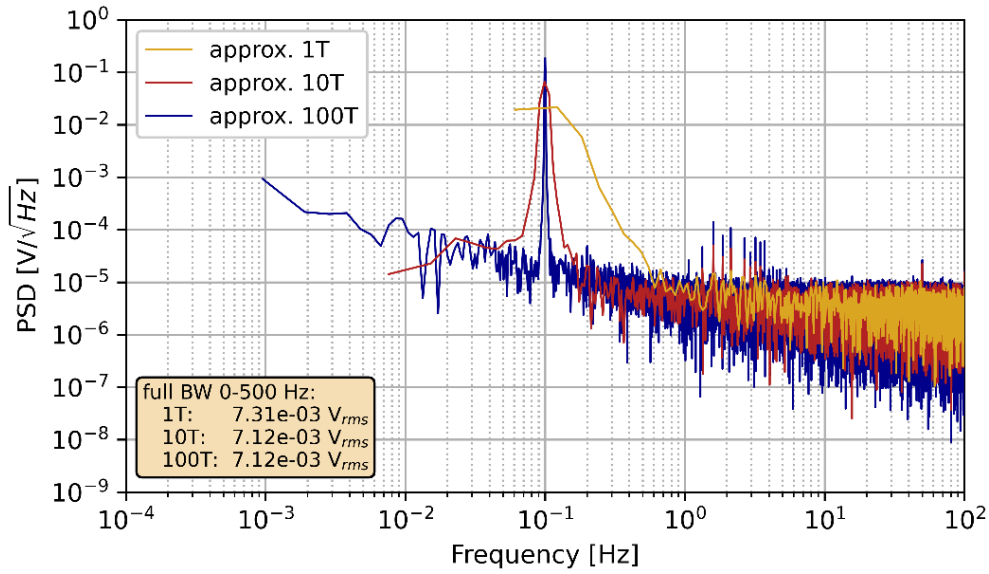


Figure 2.18 – DAQ signal power spectral density for various test durations

2.6.2 Acceleration

To couple mechanical excitation levels to dissipation induced by microvibrations, we must precisely measure acceleration levels. More specifically, we wish to measure injected acceleration (e.g. at an interface) as opposed to output acceleration (of the moving part of the test object), even if the latter is also measured sometimes during our tests. Spacecraft level specifications are also often specified in terms of *interface* acceleration levels.

Several types of accelerometer technologies exist, ranging from MEMS to magnetic induction or piezoelectric. As material properties vary with temperature and often more so at cryogenic temperatures, accelerometers must be qualified to function in a cryogenic environment. Accelerometers are placed as close to the test object as possible: either on the object itself or on the cold plate if the object is too small.

To our knowledge, there are 3 commercial accelerometer models that have demonstrated operation in low temperature environments below 4.2 K, despite being qualified for higher temperatures. All 3 are uniaxial piezoelectric-based accelerometers and are manufactured by Endevco, Kistler and PCB.

The Endevco 7703A-1000 was used during test phases of the Planck mission. This was communicated privately by IAS (Institut Astrophysique Spatial). These accelerometers were used at 2 K, despite only being qualified down to -55°C/218 K. Having charge outputs, good electromagnetic compatibility (EMC) through coaxial cables is needed to limit electromagnetic interference (EMI) and guarantee measurement chain performance. Despite charge output sensors having no dissipation (of interest in cryogenic environments), the need for EMC coaxial cables adds complexity for using them in the existing cryostats.

The Kistler 8712B5D0CB on the other hand uses integrated charge converters, the so called Integrated Electronics Piezo-Electric (IEPE) technical standard. This accelerometer is qualified down to 77 K but has

successfully been used down to 2 K by SRON (Netherlands Institute for Space Research) during the testing of their FPA for the Athena/X-IFU instrument.

Lastly, the PCB J351B41 is also qualified down to 77 K but has been used in temperatures as low as 1 K by CEA/DSBT and CNRS (E. Collin). SRON studied the impact of this lower temperature on the noise floor and gain drop. The accelerometer was found to be slightly noisier at lower temperatures and showed a gain drop of about 15% at 4.2 K compared to 77 K.²⁷ For all test campaigns of this thesis however, there was assumed to be no drop in gain, given that we are mostly studying orders of magnitude.

The PCB accelerometer is also an IEPE-based sensor, the PCB proprietary name being Integrated Circuit Piezoelectric (ICP). Having voltage output facilitates installation and reduces the risk of EMI in the downstream measurement chain. CEA/DSBT is familiar with this accelerometer and its associated signal conditioners. As discussed next, they also fulfil the desired performance criteria (temperature range aside) for accomplishing the work done during this thesis. The PCB accelerometer was therefore selected for all acceleration measurements of the cryogenic test campaigns of this thesis. It is shown in Figure 2.19.



Figure 2.19 – Cryogenic accelerometer (PCB J351B41)

Its approximate dimensions are 1.9 cm (diameter) x 2.2 cm (height). Being an ICP sensor its output is a voltage, having a sensitivity in volts per g. Some key functional specifications are listed in Table 2.9.

Table 2.9 – Key functional specifications of PCB J351B41 cryogenic accelerometer

Name	Value
<i>Sensitivity (deviation)</i> (300 K to 77 K)	100 mV/g (~5%)
<i>Resonance</i>	> 15 kHz
<i>Frequency range</i>	1-2000 Hz
<i>Temperature range</i>	77 K to 394 K (-196 °C to + 121 °C)
<i>Measurement range</i>	± 15 g _{pk}
<i>Spectral noise (0-1 Hz)</i>	100 µg/√Hz
<i>Spectral noise (1-10 Hz)</i>	22 µg/√Hz
<i>Spectral noise (10-100 Hz)</i>	6 µg/√Hz
<i>Spectral noise (100-1000 Hz)</i>	1.6 µg/√Hz

²⁷ Tests carried out at SRON thanks to Henk van Weers (private communication)

We observe that the specifications of the accelerometer correspond to the frequency range and amplitude range of microvibrations, typically 1 Hz to 1 kHz and between 10 μg to 10 mg respectively.

The accelerometers are powered using 4-channel signal conditioners (SC), namely the PCB 48205 and PCB 482C15. The first is a fixed gain SC (x1) and the second has an adjustable gain (x1, x10 or x100). Their role is also to linearize the voltage signals, ensuring a constant sensitivity (volts per g). The output of the SCs is then fed to the voltage measurement system. The SC external casing (the same for both SC models) is shown in Figure 2.20.



Figure 2.20 – Signal conditioner (PCB 482CXX) front panel (left) and rear panel (right)

It is interesting to compare noise floors of the accelerometer and SC models used. Key spectral noise and frequency response specifications of the accelerometers (identical to those presented in Table 2.9) and SCs is shown in Table 2.10.

Table 2.10 – Key specifications of PCB cryogenic accelerometer and signal conditioners

	Acc. (J351B41)	SC (482C05)	SC (482C15) (x1)	SC (482C15) (x100)
<i>Low freq. response</i>	-	< 0.1 Hz	< 0.05 Hz	< 0.05 Hz
<i>High freq. response.</i>	-	> 1 kHz	> 100 kHz	> 17 kHz
<i>Spectral noise (1 Hz)</i>	10 $\mu\text{V}/\sqrt{\text{Hz}}$	1.30 $\mu\text{V}/\sqrt{\text{Hz}}$	0.67 $\mu\text{V}/\sqrt{\text{Hz}}$	57 $\mu\text{V}/\sqrt{\text{Hz}}$
<i>Spectral noise (10 Hz)</i>	2.2 $\mu\text{V}/\sqrt{\text{Hz}}$	0.10 $\mu\text{V}/\sqrt{\text{Hz}}$	0.10 $\mu\text{V}/\sqrt{\text{Hz}}$	5.2 $\mu\text{V}/\sqrt{\text{Hz}}$
<i>Spectral noise (100 Hz)</i>	0.6 $\mu\text{V}/\sqrt{\text{Hz}}$	0.08 $\mu\text{V}/\sqrt{\text{Hz}}$	0.06 $\mu\text{V}/\sqrt{\text{Hz}}$	1.7 $\mu\text{V}/\sqrt{\text{Hz}}$
<i>Spectral noise (1000 Hz)</i>	0.16 $\mu\text{V}/\sqrt{\text{Hz}}$	0.07 $\mu\text{V}/\sqrt{\text{Hz}}$	0.06 $\mu\text{V}/\sqrt{\text{Hz}}$	1.8 $\mu\text{V}/\sqrt{\text{Hz}}$
<i>Spectral noise (10 000 Hz)</i>	-	0.07 $\mu\text{V}/\sqrt{\text{Hz}}$	0.05 $\mu\text{V}/\sqrt{\text{Hz}}$	1.4 $\mu\text{V}/\sqrt{\text{Hz}}$

The spectral noise levels of the accelerometer provided in $\mu\text{g}/\sqrt{\text{Hz}}$ (shown in Table 2.9), these are converted to $\mu\text{V}/\sqrt{\text{Hz}}$ (shown in Table 2.10) for ease of comparison to the SC spectral noise levels. This is done using the sensitivity of 100 mV/g.

The 2 rightmost columns show the specifications for the gain-adjustable SC at a gain of 1 and 100. For a gain of 100, the signal noise floor originating from the accelerometers is amplified by a factor 100 too. This means that column 1 (leftmost) of Table 2.10 can directly be compared to columns 2 and 3, whereas a factor 100 must be used to compare columns 1 and 4. Either way, we note from Table 2.10 that the noise floor of the full measurement chain (accelerometer + SC) is dominated by the accelerometer

spectral noise levels. This is particularly true when comparing the noise levels of the accelerometer and SC set to a gain of 100. Across the 100 Hz band for example, their respective contributions are $60 \mu\text{V}/\sqrt{\text{Hz}}$ and $1.7 \mu\text{V}/\sqrt{\text{Hz}}$. This corresponds to acceleration noise levels of $6 \mu\text{g}/\sqrt{\text{Hz}}$ and $0.17 \mu\text{g}/\sqrt{\text{Hz}}$ respectively.

These theoretical acceleration noise floors will allow us to later estimate the full measurement chain noise floor, from which we will be able to deduce a theoretical lower limit of the observable acceleration level.

Having now discussed acceleration measurement and its associated noise, we will now cover the topic of displacement measurement. The full measurement chain noise from sensor (i.e. accelerometer or displacement sensor) to voltage measurement will then be presented.

2.6.3 Displacement

In addition to the 2 quantities measured at cryogenic temperatures (i.e. acceleration and temperature), displacement measurement is also of interest. Several types of non-contact displacement sensors exist, such as capacitive, ultrasound-based or laser-based. Those used are laser sensors: the Keyence IL-065 and the Keyence LK-G82. Both sensors are based on laser triangulation, whereby a laser is emitted and reflected by the test object onto a sensor head. The angle of the reflected light to the incident light is proportional to the displacement of the test object with respect to the laser sensor. When the reflecting surface shows high rotation or bending, this can therefore introduce a slight error in the laser reading. Such commercial sensors are only qualified for ambient temperatures however, not for cryogenic environments. The resolution and noise levels of both laser sensors are compared in Table 2.11.

Table 2.11 – Key specifications of displacement sensors

	Keyence IL-065	Keyence LK-G82
<i>Sensitivity</i>	500 mV/mm	1000 mV/mm
<i>Measurement range</i>	± 10 mm	± 15 mm
<i>Reference distance</i>	65 mm	80 mm
<i>Repeatability (precision)</i>	2 μm	0.2 μm
<i>Sampling frequency (default)</i>	1 kHz	1 kHz
<i>Sampling frequency (max)</i>	up to 3 kHz	up to 50 kHz
<i>Operation temperature</i>	-10 to +50 °C	0 to +50 °C

The laser sensors are used to determine resonance frequencies of a test object, given it is not enclosed and inaccessible by laser pointing. Some test objects studied in this thesis are small enough for the addition of an accelerometer to impact its resonance frequency. Laser sensors allow to measure said frequency without changing it. Resonance frequencies have been found to slightly shift between ambient temperature and cryogenic temperatures, as we shall see later. However, determining the resonance frequency (even if approximate to within a few Hz) using said laser sensors still informs use on which spectral regions to explore when studying microvibration dissipation at cryogenic temperatures.

Another use of the laser sensor is to measure the displacement of interfaces (e.g. the cold plate or support bracket) to evaluate the soundness of the mechanical environment in which we are working.

To validate the use of these laser sensors for studying microvibrations (namely regarding typical amplitude and frequency ranges), we compare their respective noise levels. The lasers are pointed on a metal block resting on an optical work bench. The noise spectral densities of these tests are shown in Figure 2.21.

The measurements have a duration of approximately 21 s at a sampling rate of 100 kHz. This leads to a theoretical minimum frequency of 95 mHz ($2 \times 1/21$) and a Nyquist frequency of 50 kHz, having applied Welch segment lengths of $T/2$ and 50% overlap. From the figure we also note the broadband noise levels, as well as the spectral noise levels at 100 Hz and 1000 Hz, all of which are summarised in Table 2.12.

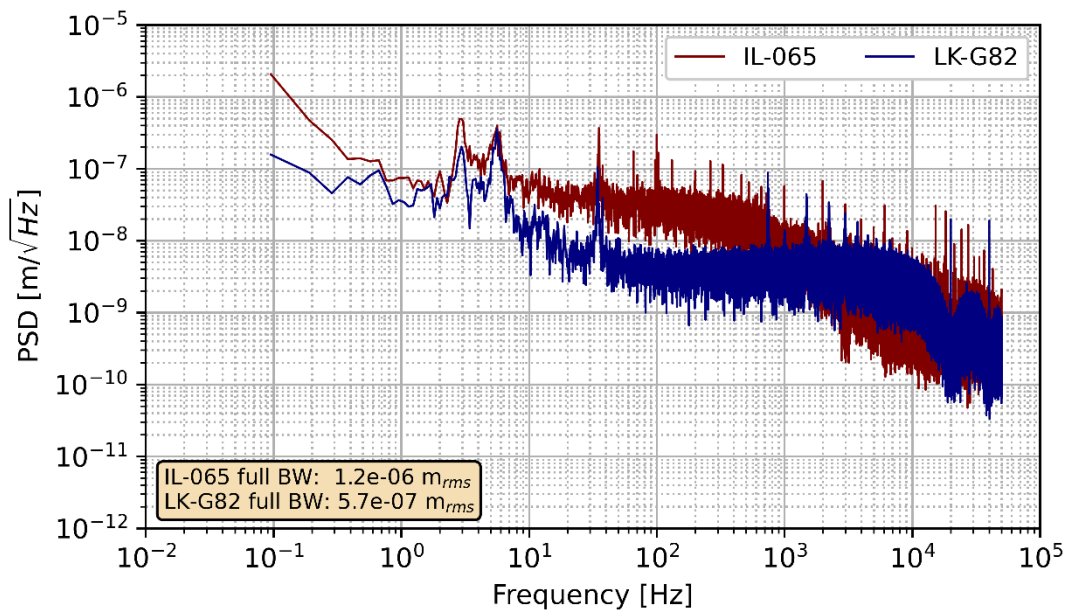


Figure 2.21 – Noise spectral densities of laser displacement sensors at 300 K

The broadband noise of the IL-065 sufficing, all displacement measurements presented in the remainder of this thesis have used this laser.

Table 2.12 – Measured noise levels of laser displacement sensors

Name	Keyence IL-065	Keyence LK-G82
<i>Broadband noise (0-50 kHz)</i>	1.2 μm_{rms}	0.6 μm_{rms}
<i>Spectral noise (100 Hz)</i>	300 nm/ $\sqrt{\text{Hz}}$	8 nm/ $\sqrt{\text{Hz}}$
<i>Spectral noise (1000 Hz)</i>	60 nm/ $\sqrt{\text{Hz}}$	20 nm/ $\sqrt{\text{Hz}}$

Having now presented the components of the mechanical measurement chains separately, we can cover the full measurement chain noise, from sensor to voltage measurement.

2.6.4 Full chain noise

Given the voltage measurements systems (i.e. DAQ and spectrum analyser) and corresponding sensors (i.e. accelerometers and displacement sensors), we can estimate the full measurement chain noise levels. We first summarise the noise levels of each component of the measurement chain in Table 2.13. These are obtained either from the measured noise levels or from the system specifications previously discussed, and are at the reference frequencies of 100 Hz and 1 kHz. The mechanical noise floors are translated to voltage noise floors using the sensor sensitivities: 100 mV/g and 500 mV/mm for the accelerometer and chosen laser respectively.

From these noise levels we deduce that when the gain is set to 1, the accelerometer and DAQ system are the limiting components of the acceleration measurement chain over the 100-1000 Hz range (at $0.6 \mu\text{V}/\sqrt{\text{Hz}}$ and $0.5 \mu\text{V}/\sqrt{\text{Hz}}$ respectively). When setting the gain to 100 however, the accelerometer becomes the main limiting component with regards to noise (now at $60 \mu\text{V}/\sqrt{\text{Hz}}$). For the most part, the tests of this thesis (made in the typical microvibration range of 1 Hz to 1 kHz), used a gain of 100.

Table 2.13 – Voltage noise spectral densities (at 100 Hz and 1 kHz) for all mechanical equipment

	Noise 100 Hz	Noise at 1000 Hz	Deduced from
<i>Accelerometer</i>	$0.6 \mu\text{V}/\sqrt{\text{Hz}}$	$0.16 \mu\text{V}/\sqrt{\text{Hz}}$	Table 2.10
<i>Laser sensor (IL-065)</i>	$150 \mu\text{V}/\sqrt{\text{Hz}}$	$30 \mu\text{V}/\sqrt{\text{Hz}}$	Table 2.12
<i>SC (482C15) (x1)</i>	$0.06 \mu\text{V}/\sqrt{\text{Hz}}$	$0.06 \mu\text{V}/\sqrt{\text{Hz}}$	Table 2.10
<i>DAQ</i>	$0.5 \mu\text{V}/\sqrt{\text{Hz}}$	$0.5 \mu\text{V}/\sqrt{\text{Hz}}$	Table 2.7
<i>SA</i>	$5 \text{ nV}/\sqrt{\text{Hz}}$	$5 \text{ nV}/\sqrt{\text{Hz}}$	Table 2.8

We can now estimate the spectral noise floors of each full measurement chain, from a combination of equipment specifications and measured values. Two variations of the acceleration measurement chain exist, using either the DAQ system or spectrum analyser for voltage measurement. Summing the spectral noise levels for each measurement chain, at 100 Hz and 1000 Hz, yields Table 2.14. Voltage levels are translated back to equivalent mechanical levels, again using the sensor sensitivities.

Table 2.14 – Estimated acceleration and displacement noise levels (at 100 Hz and 1 kHz) for SC gains 1 and 100

	Measurement chain	Noise at 100 Hz		Noise at 1000 Hz	
		Gain x1	Gain x100	Gain x1	Gain x100
<i>Acceleration (w/ DAQ)</i>	Acc. + SC + DAQ	$11.6 \mu\text{g}/\sqrt{\text{Hz}}$	$6.7 \mu\text{g}/\sqrt{\text{Hz}}$	$7.2 \mu\text{g}/\sqrt{\text{Hz}}$	$2.3 \mu\text{g}/\sqrt{\text{Hz}}$
<i>Acceleration (w/ SA)</i>	Acc. + SC + SA	$6.7 \mu\text{g}/\sqrt{\text{Hz}}$	$6.6 \mu\text{g}/\sqrt{\text{Hz}}$	$2.3 \mu\text{g}/\sqrt{\text{Hz}}$	$2.2 \mu\text{g}/\sqrt{\text{Hz}}$
<i>Displacement (IL-065)</i>	Laser + DAQ or SA	$\sim 300 \text{ nm}/\sqrt{\text{Hz}}$		$\sim 60 \text{ nm}/\sqrt{\text{Hz}}$	

Bearing in mind that these are ballpark figures, they nevertheless provide an indication of the capabilities of the full acceleration and displacement measurement chains. We also note that using a gain of 100 lowers the acceleration noise floors, particularly when using the DAQ system. Again, this is because for

a gain set to 1, of all components in the acceleration measurement chain the DAQ system noise contribution is greatest at $0.5 \mu\text{V}/\sqrt{\text{Hz}}$ (see Table 2.13). The noise of the spectrum analyser is only $5 \text{ nV}/\sqrt{\text{Hz}}$ though, thus making little difference to the total noise floor when the gain is set to 100.

Overall the mechanical measurement chain components are found to be well suited for the study of microvibrations in the typical ranges discussed in section 2.4, namely 1 Hz to 1 kHz and $10 \mu\text{g}$ to 10 mg .

Having now covered mechanical sensing, we will now discuss mechanical actuation.

2.6.5 Mechanical excitation

To induce controlled mechanical vibrations, a small mechanical shaker is used. The shaker used is a TMS K2007E01 sold by PCB. It is an electrodynamic shaker used to induce microvibrations in the test objects studied. As it too is not qualified for cryogenic environments, microvibrations are induced from the cryostat's exterior (see Figure 2.1) and travel through it to reach the cold plate and test object. The shaker is suspended and excites the cryostat via a $\sim 15 \text{ cm}$ long nylon rod and interface plate, shown Figure 2.22.

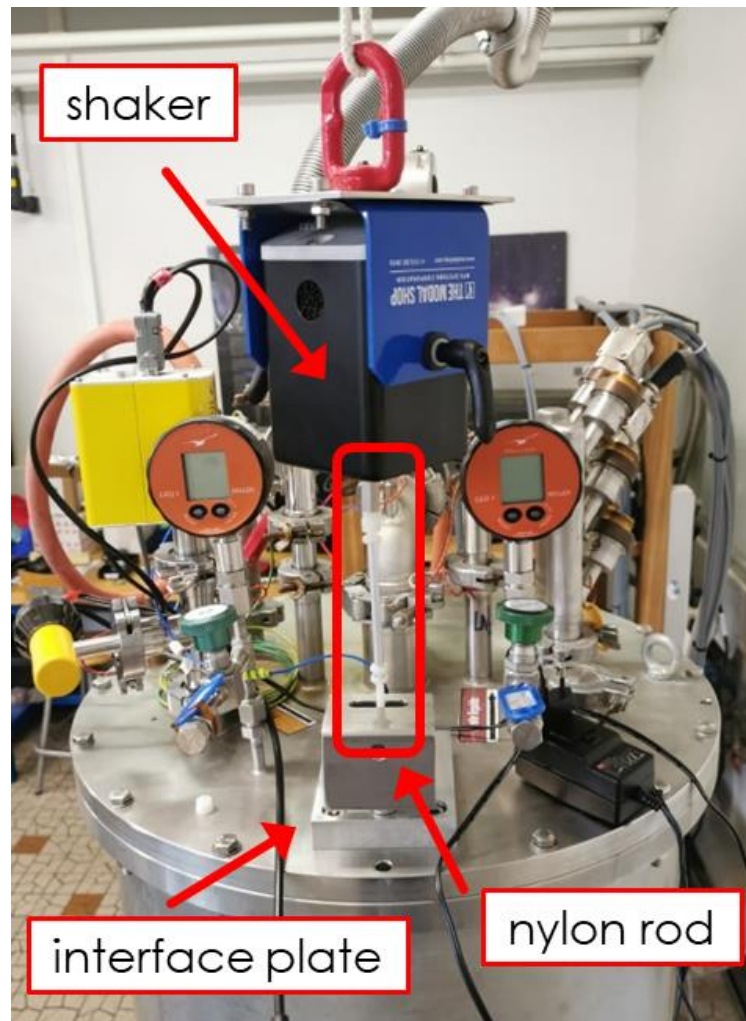


Figure 2.22 – Mechanical shaker with nylon rod as interface with cryostat exterior

Some key specifications of the mechanical shaker are summarised in Table 2.15.

Table 2.15 – Key functional specifications of mechanical shaker

Name	Value
<i>Stroke</i>	13 mm
<i>Output force (sine peak)</i>	31 N
<i>Output force (random RMS)</i>	22 N
<i>Frequency range</i>	DC to 9 kHz
<i>Distortion (at 1 kHz, 1 W)</i>	< 0.02 %
<i>Gain level 1</i>	10 dB
<i>Gain level 2</i>	18 dB
<i>Gain level 3</i>	24 dB

For the study of microvibrations, the shaker's frequency range and distortion are crucial specifications. Specifically, it is key that frequencies other than the command frequency are not excited, to correctly deduce the excitation frequencies that cause microvibration dissipation for a given system. The typical range of microvibrations (i.e. 10 Hz to 1 kHz) falls well within the shaker's excitation range. In combination with its very low harmonic distortion, this mechanical shaker is ideal for studying the effects of specific mechanical frequencies on the dissipation of cryogenically cooled systems.

The shaker is controlled by an AC voltage provided by an arbitrary function generator (AFG). It generates constant (e.g. fixed amplitude and fixed frequency) or time-varying signals (e.g. frequency sweeps or amplitude modulation), used to induce microvibrations in the test object. A Tektronix AFG3022B is used.

2.6.5.1 Shaker harmonic distortion

Harmonic distortion was studied by mounting the shaker in conjunction with an accelerometer on a vibration-isolated optical bench, shown in Figure 2.23 and Figure 2.24.



Figure 2.23 – Mechanical shaker mounted on optical bench & accelerometer mounted on shaker moving part

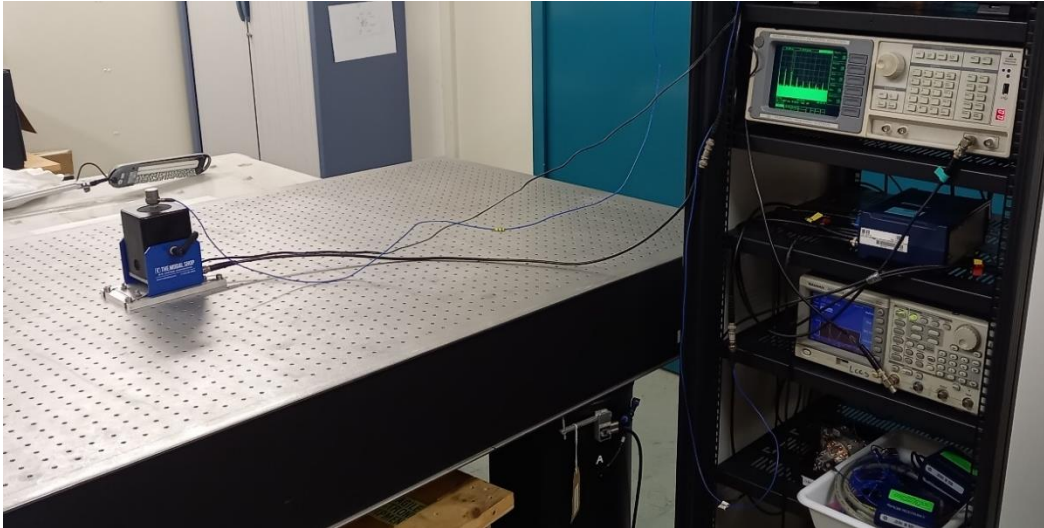


Figure 2.24 – Mechanical shaker mounted on optical bench and test rack

For a given excitation frequency, total harmonic distortion (THD) was recorded using the spectrum analyser. The test plan adopted followed recommendations of the shaker user manual:

- adjust excitation voltage for a 2 g response at 100 Hz
- sweep from 10 Hz to 10 kHz at this fixed excitation voltage

The reference excitation of 2 g at 100 Hz was found to correspond to a shaker drive voltage of 18.5 mV_{rms} at a gain level of 1 (see Table 2.15). This gain level is used for the entirety of this thesis. The resulting broadband acceleration measurements are shown in Figure 2.25. It is worth noting the peak at 7 kHz, slightly below the stated maximum shaker operation frequency. Within our target frequency band though (10 Hz to 1 kHz), the shaker response at fixed voltage spans roughly one order of magnitude.

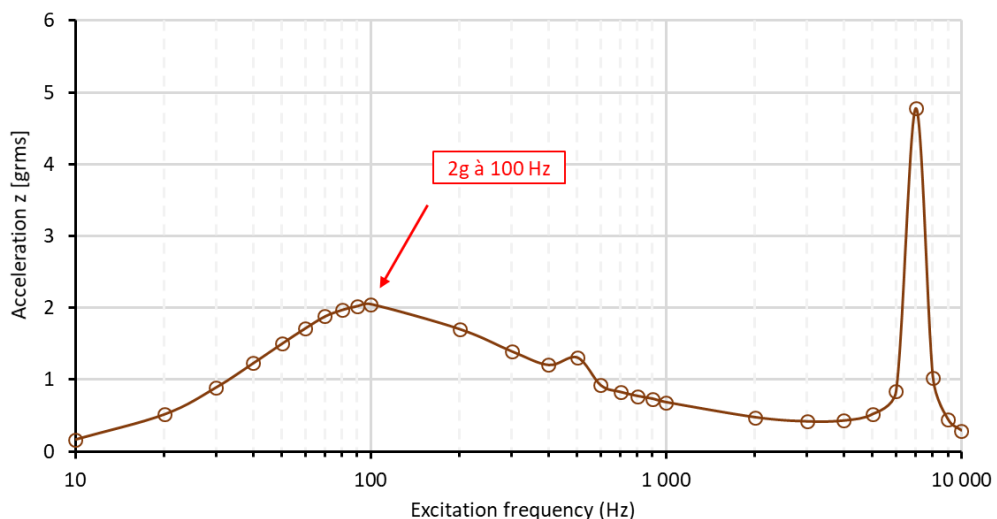


Figure 2.25 – Measured broadband acceleration of mechanical shaker

The corresponding THD values for each excitation frequency is shown in Figure 2.26.

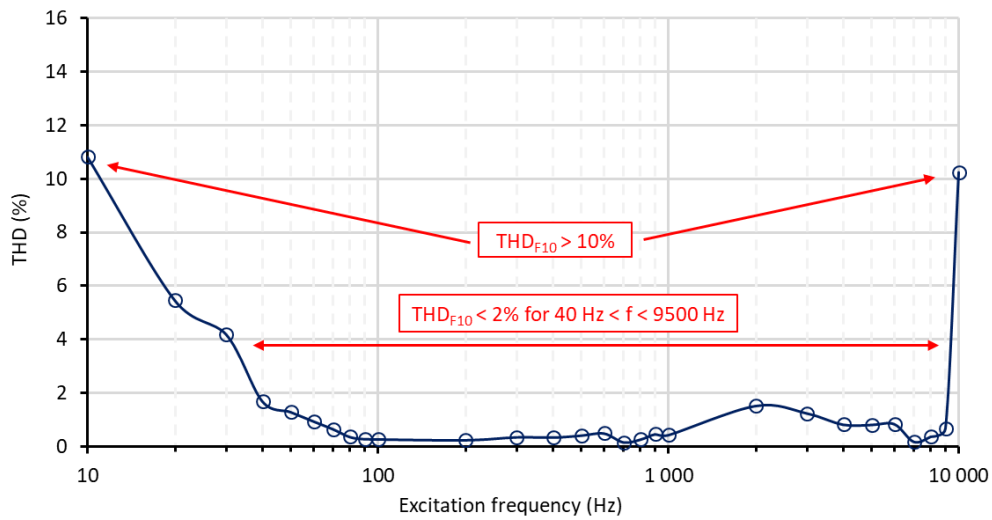


Figure 2.26 – Harmonic distortion of shaker as measured by accelerometer and spectrum analyser

We note that harmonic distortion does not exceed 2% for an excitation frequency between 40 Hz and 9.5 kHz. Beyond this frequency band distortion rises to around 10%. Though this could be problematic in the 10 Hz to 100 Hz range given the typical microvibration ranges discussed section 2.4, the system studied in this thesis have in fact displayed resonances between 100 Hz and 500 Hz. In this range distortion does not exceed 1%, making this mechanical shaker well-suited for our studies.

2.6.6 Summary

To summarise this section on mechanical sensing and actuation, the schematic of the microvibration cryostat test bench is shown again in Figure 2.27. Here we have covered all the components in blue:

- **Mechanical actuation** through the function generator that commands the mechanical shaker.
- **Mechanical sensing** through the cryogenic accelerometers that are read by the DAQ system and spectrum analyser via the signal conditioner.

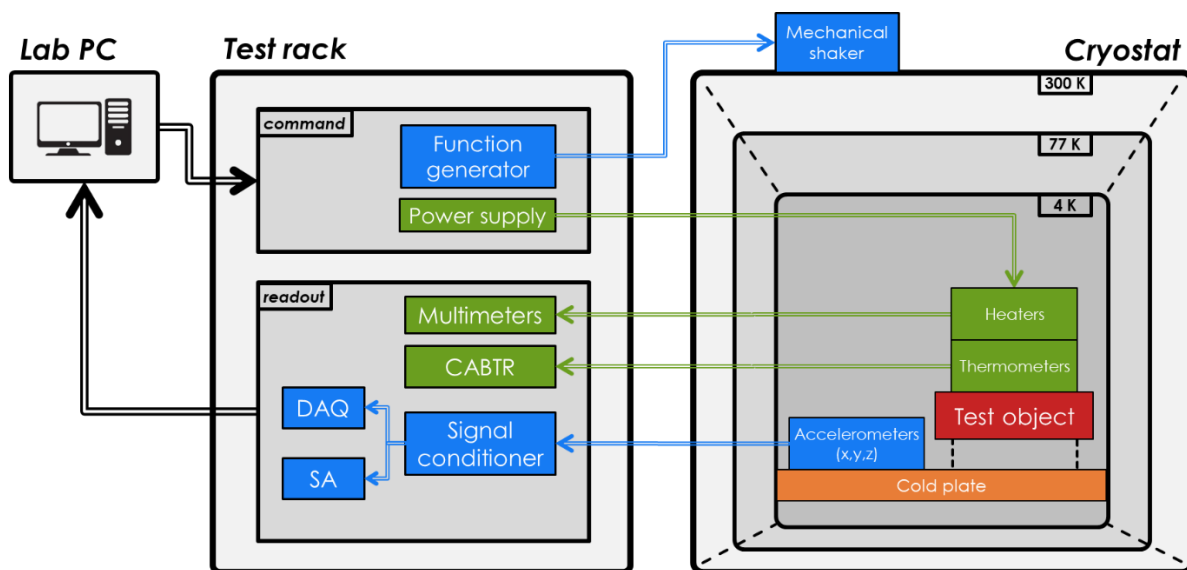


Figure 2.27 – Schematic of microvibration cryostat test bench (cold configuration)

Details on the ranges and smallest acceleration levels that can be resolved with the discussed equipment will be covered in the conclusion of this chapter, after covering the thermal equipment used.

2.7 Thermal sensing and actuation

Regarding thermal sensing and actuation used to study microvibration dissipation at low temperatures, this section is divided into 2 topics: temperature and power.

2.7.1 Temperature

This section on temperature measurement will first present the cryogenic thermometers used, followed by the associated readout electronics for low-temperature measurement.

2.7.1.1 Thermometers

The thermometers used for our experiments are 4-wire Lake Shore Cernox CX-1050-CU thermometers, shown in Figure 2.28. Their relatively small size (approximately $\varnothing 8$ mm x 4 mm) allows them to be placed almost anywhere in and around the test object, using an M3 screw. The temperature is deduced from the measurements of their electrical resistance.

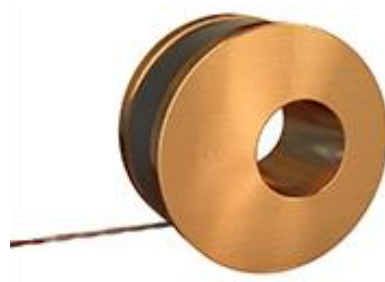


Figure 2.28 – Cryogenic thermometer (Cernox CX-1050-CU)

The thermometers are calibrated by Lake Shore between ~ 300 K and ~ 1.4 K, allowing precise temperature readouts in this range. After calibration, their accuracy at 4.2 K and 1.2 K is ± 5 mK. More importantly for our experiments, the temperature noise level is dependent on the noise of the temperature acquisition electronics. As we shall see, the lower this noise level, the more finely we are able to measure the thermal impact of microvibration dissipation.

2.7.1.2 Temperature readout electronics

The readout electronics used with the low-temperature thermometers is an in-house solution, the so-called *Centrale d'Acquisition Basses Températures Rapide* (CABTR). This readout system can read up to 8 channels and has a bandwidth of up to 100 Hz. The precision and accuracy of the CABTR system's temperature measurement depends on the resistance of the thermometer used. Some key measurement specifications of the CABTR system, for given thermometer resistances, are shown in Table 2.16.

Accuracy refers to how close a given set of measurements are to the true value, in other words that their mean is close to the true value. Precision on the other hand, refers to how close these measurements are to each other, irrespective of the true value. The noise of a signal refers to precision rather than accuracy.

For our work, we are more interested in precision than accuracy, as a measurement offset (reduced accuracy) does not noticeably affect our tests, given the measurement methods we employ (these will be presented in chapter 3). High precision on the other hand (i.e. lower noise), increases the resolution of our measurements (i.e. the smallest temperature or dissipation difference we are able to distinguish).

We quantify the precision using the coverage factor k that indicates the confidence level that data points lie within a given range. It is expressed as the number of standard deviations: a confidence level of 68% ($k=1$), of 95% ($k=2$) or 99.7% ($k=3$). The rightmost column of Table 2.16 is for $k=2$.

Table 2.16 – Key measurement specifications of CABTR system

<i>Thermometer resistance</i>	<i>Accuracy</i>	<i>Precision ($k=2$)</i>
1 Ω to 5 Ω	$\pm 0.002 \Omega \pm 0.05\% R$	0.07% R
5 Ω to 10 Ω	$\pm 0.002 \Omega \pm 0.05\% R$	0.02% R
10 Ω to 50 Ω	$\pm 0.002 \Omega \pm 0.01\% R$	0.01% R
50 Ω to 100 Ω	$\pm 0.01\% R$	0.005% R
100 Ω to 500 Ω	$\pm 0.01\% R$	0.002% R
500 Ω to 7 k Ω	$\pm 0.01\% R$	0.002% R
7 k Ω to 30 k Ω	$\pm 0.01\% R$	0.005% R
30 k Ω to 100 k Ω	$\pm 0.04\% R$	0.02% R

From the specifications listed in Table 2.16, and the resistances and sensitivities of our thermometers, we deduce the equivalent *temperature* accuracy and noise (i.e. precision). These are given in Table 2.17.

Table 2.17 – Temperature noise and accuracy due to CABTR system

<i>Thermometer</i>	<i>Temperature</i>	<i>Resistance</i>	<i>Sensitivity</i>	<i>Temperature Accuracy</i>	<i>Temperature Noise ($k=2$)</i>
#1 (X149831)	300 K	62 Ω	-0.187 Ω/K	± 33 mK	17 mK
	77 K	210 Ω	-2.287 Ω/K	± 9 mK	1.8 mK
	4.2 K	2300 Ω	-580.7 Ω/K	± 400 μK	80 μK
	1.2 K	16 500 Ω	-26885 Ω/K	± 60 μK	31 μK
#2 (X90579)	300 K	39 Ω	-0.086 Ω/K	± 69 mK	46 mK
	77 K	94 Ω	-0.329 Ω/K	± 13 mK	7 mK
	4.2 K	390 Ω	-50.92 Ω/K	± 780 μK	160 μK
	1.2 K	960 Ω	-745.0 Ω/K	± 130 μK	27 μK
#3 (X84246)	300 K	58 Ω	-0.123 Ω/K	± 47 mK	24 mK
	77 K	132 Ω	-0.928 Ω/K	± 14 mK	2.8 mK
	4.2 K	571 Ω	-83.30 Ω/K	± 690 μK	140 μK
	1.2 K	1658 Ω	-1543 Ω/K	± 110 μK	21 μK

The temperature noise of the rightmost column only pertains to the electrical noise from the readout system. Other sources of temperature measurement fluctuations, say fluctuations in the bath temperature (as we shall see in chapter 3) or electromagnetic interference in the cables, are not included here.

We note that this theoretical noise falls sharply with temperature, reaching a level no greater than 30 μK at 1.2 K. This is a key figure for our experiments and will be compared to experimental values.

2.7.2 Power

This section on power measurement will first present the equipment used, namely multimeters, a power supply and resistive heaters. Then a comparison of 2 power measurement methods will be made, before covering an experimental approach to improving the power measurement precision.

2.7.2.1 Heaters and equipment

Electrical heaters are used throughout this thesis, either for calibration or thermal regulation in and around the test object. Those used are Vishay resistors, either 402 Ω or 5000 Ω which are metal oxide film resistors. We shall name these R_1 and R_2 respectively. They are attached to a holed copper fixing plate using Stycast, an epoxy showing relatively high thermal conductivity. This is shown in Figure 2.29.



Figure 2.29 – Electrical heater (Vishay) with copper fixing plate

Some key specifications of these resistors are summarised in Table 2.18. Their electrical resistance is almost fully independent of temperature, with a temperature coefficient of resistance of 5 ppm/°C.

Table 2.18 – Key specifications of electrical heaters

<i>Name</i>	<i>Value</i>
<i>Resistance</i>	402 Ω or 5 k Ω
<i>Resistance tolerance</i>	$\pm 0.1\%$
<i>Temperature Coefficient of Resistance</i>	5 ppm/°C
<i>Power rating</i>	125 mW

Though this change in resistance is only stated down to -55°C (218 K), these heaters are known to have almost no change even down to very low temperatures. A quick test was carried out to check the variation from 300 K to 77 K by submerging the heaters in liquid nitrogen. Changes of 0.17% and 0.16% were found for R_1 and R_2 respectively. Known to be constant down to 50 mK, we can assume that the heaters have a fixed resistance down to 1.2 K (our lowest working temperature). This is all the more true at a given working temperature.

The most accurate way to measure joule heating is through the robust 4-wire method, whereby one pair of leads is used to inject current and another pair for sensing voltage. Joule heating is thus deduced solely from the injected current and measured voltage at resistance level, therefore independently of the resistance value. Simpler but less reliable methods are based on either voltage or current measurement (not both), assuming constant electrical resistance.

As we have assumed that the resistances of R_1 and R_2 are fixed, deducing joule heating by current measurement alone (henceforth called “current method”) should be sufficiently accurate. A verification of the accuracy of the current method versus the 4-wire method is done below, using the specifications of the power supply and multimeters used.

The multimeter used is the FI 3136A. Some of its key specifications are shown in Table 2.19.

Table 2.19 – Key specifications of multimeter (FI 3136A)

Name	Value
<i>DC current range (min)</i>	500 μ A
<i>DC voltage range (min)</i>	500 mV
<i>DC current resolution (max)</i>	10 nA
<i>DC voltage resolution (max)</i>	10 μ V

The power supply used is an Agilent E3647A and some of its key specifications are shown in Table 2.20. One of the most crucial of these for our experiments is the voltage command resolution of 1 mV.

Table 2.20 – Key specifications of power supply (Agilent E3647A)

Name	Value
<i>Maximum power</i>	60 W
<i>DC output (1 channel)</i>	60 V / 0.5 A
<i>DC output (2 channels)</i>	35 V / 0.8 A
<i>Voltage command stability (over 8 hours)</i>	< 0.02% + 2 mV
<i>Voltage command resolution</i>	< 5 mV (spec), 1 mV (exp.)

2.7.2.2 Measurement methods

From the multimeter specifications (both for current and voltage readout), we can deduce theoretical power readout error (i.e. noise) for both the current method and the 4-wire method. For a function f , having variables $x_1, x_2, \dots, x_n$ that themselves have uncertainties $\Delta x_1, \Delta x_2, \dots, \Delta x_n$, error propagation states that the error Δf is given by:

$$\Delta f = \sqrt{\left(\frac{\partial f}{\partial x_1}\right)^2 \Delta x_1^2 + \left(\frac{\partial f}{\partial x_2}\right)^2 \Delta x_2^2 + \dots + \left(\frac{\partial f}{\partial x_n}\right)^2 \Delta x_n^2} \quad (2.3)$$

We shall take 2 reference power levels that represent the range within which we wish to observe microvibration dissipation: 100 nW and 10 μ W. Table 2.21 summarises the noise in electrical power readout for each of these power levels, for both methods and both heaters. The ΔI and ΔV values correspond to the multimeter readout resolutions, given in the last 2 rows of Table 2.19.

At a given working temperature, we can reasonably assume that the heaters have a constant resistance, as discussed in section 2.7.2.1. Furthermore, the Johnson is negligible in our case. For the current method we therefore take $\Delta R \approx 0$, simplifying the power error formula.

Table 2.21 – Power readout noise (theoretical)

Name		Current measurement	4-wire
Power function		$P = RI^2$	$P = VI$
Power error formula		$\Delta P = RI^2 \sqrt{\left(\frac{\Delta R}{R}\right)^2 + 4\left(\frac{\Delta I}{I}\right)^2} \approx 2RI\Delta I$	$\Delta P = VI \sqrt{\left(\frac{\Delta V}{V}\right)^2 + \left(\frac{\Delta I}{I}\right)^2}$
Readout noise $R_1 = 402 \Omega$	100 nW	0.13 nW	0.17 nW
	10 μ W	1.3 nW	1.7 nW
Readout noise $R_2 = 5000 \Omega$	100 nW	0.45 nW	0.23 nW
	10 μ W	4.5 nW	2.3 nW

We note that for the dissipation range we wish to study, the heaters chosen and the multimeter chosen, power readout noise does not exceed 4.5 nW, irrespective of the measurement method. It must be noted however, that these are best case values, namely when the multimeter bits are concentrated over the smallest current and voltage ranges, namely 500 μ A and 500 mV respectively as per Table 2.19. Given that no significant gain in readout precision is achieved when using the 4-wire method, the current method was therefore used for power readout for the remainder of thesis.

The values of Table 2.21 only indicate power *readout* noise however, not the total power measurement precision. They only reflect noise due to multimeter readout, in the case of a perfectly stable and noiseless voltage source. In reality however, the latter is imperfect and is present irrespective of the readout method used. The stability and resolution of the voltage command given in Table 2.20 are used to estimate their corresponding contribution to said total power measurement precision.

The voltage command stability is the stability at a fixed voltage command. The voltage command resolution is the smallest voltage step achievable by the voltage source. Each is translated to power units, which we refer to as the power *command* noise. From Table 2.20 we note that the stated voltage command resolution is independent of the voltage level and at the power levels studied (100 nW and 10 μ W) the voltage command stability is very close to 2 mV, for both R_1 and R_2 . Hence the command stability and resolution values are similar for both power levels (and all those in between). The values are shown in Table 2.22.

Table 2.22 – Power command noise (theoretical)

Name	$R_1 = 402 \Omega$	$R_2 = 5000 \Omega$
Command stability	10 nW	0.8 nW
Command resolution	62 nW	5 nW

Here too the values are best case scenarios, namely when the voltage supply is operating over its smallest voltage range. We note that the greatest contributor to the total power measurement precision is the command resolution. We therefore focus on this to improve overall power measurement precision.

2.7.2.3 Measurement precision

Power measurement at fixed voltage command is shown in Figure 2.30, for the 2 heaters. We also note the impact of a 1 mV step in the fixed voltage command on the power measured. This step corresponds to the command resolution of the power supply, as per Table 2.20. The fixed voltages chosen here correspond to heating that regulate the heat switch base and tip at about 4.4 K and 6.0 K respectively.

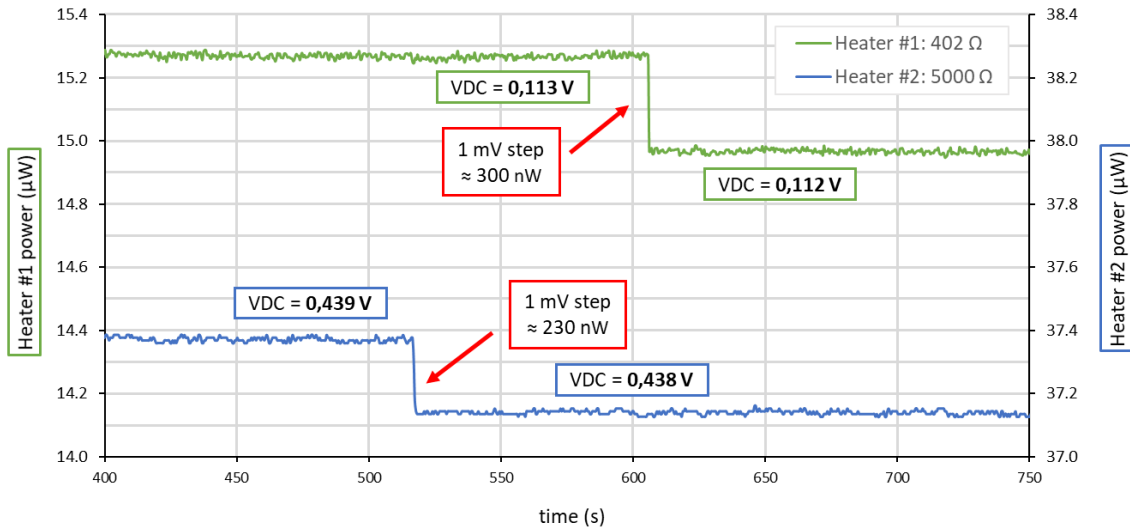


Figure 2.30 – Power measurement precision: impact of switching 1 mV step

Experimentally, a voltage step of 1 mV leads to a corresponding 300 nW and 230 nW electrical power step across R_1 and R_2 respectively. We verify these values theoretically in Table 2.23.

We first note from Figure 2.30 the measured powers across the heaters before the 1 mV step. The current through them is then deduced. Given the fixed voltage commands before the 1 mV step (i.e. 0.113 V and 0.439 V for R_1 and R_2 respectively), we then deduce the total series resistance. Lastly, we compute the step in electrical power solely across the heater, due to a 1 mV step on this total resistance.

Hence we find that a 1 mV step causes 270 nW and 170 nW steps in electrical power across R_1 and R_2 respectively. This agrees with the experimentally observed steps that are only slightly higher.

Table 2.23 – Impact of 1 mV step on heater electrical power

Name	$R_1 = 402 \Omega$	$R_2 = 5000 \Omega$
Power across heater (before 1 mV step)	15.3 μ W	37.4 μ W
Current through heater (before 1 mV step)	195 μ A	86 μ A
Total series resistance (heater + wiring)	579 Ω	5076 Ω
Power step across heater (after 1 mV step)	270 nW	170 nW

We can also verify these values analytically. Neglecting the resistance in the wires, the power step ΔP due to the voltage step ΔV solely across the heater is given by:

$$\Delta P = 2 \sqrt{\frac{P}{R_{heater}}} \Delta V \cdot \frac{R_{heater}}{R_{total}} = 2 \cdot \frac{\sqrt{P R_{heater}}}{R_{total}} \Delta V \quad (2.4)$$

For example for $R_2 = 5000 \Omega$, where the resistance in the wires is negligible, for $P = 37.4 \mu\text{W}$ and $\Delta V = 1 \text{ mV}$, we obtain $\Delta P = 173 \text{ nW}$. This is in agreement with the bottom right value of Table 2.23.

We observe that we can further improve power resolution by increasing R_{total} . Adding an external (outside of the cryostat) resistance in series divides the 1 mV step across this additional resistance and the heater. The power step ΔP across the heater is then reduced. We observe this experimentally in Figure 2.31, where the impact of the addition of 10 k Ω and 100 k Ω resistances to R_1 and R_2 respectively is shown.

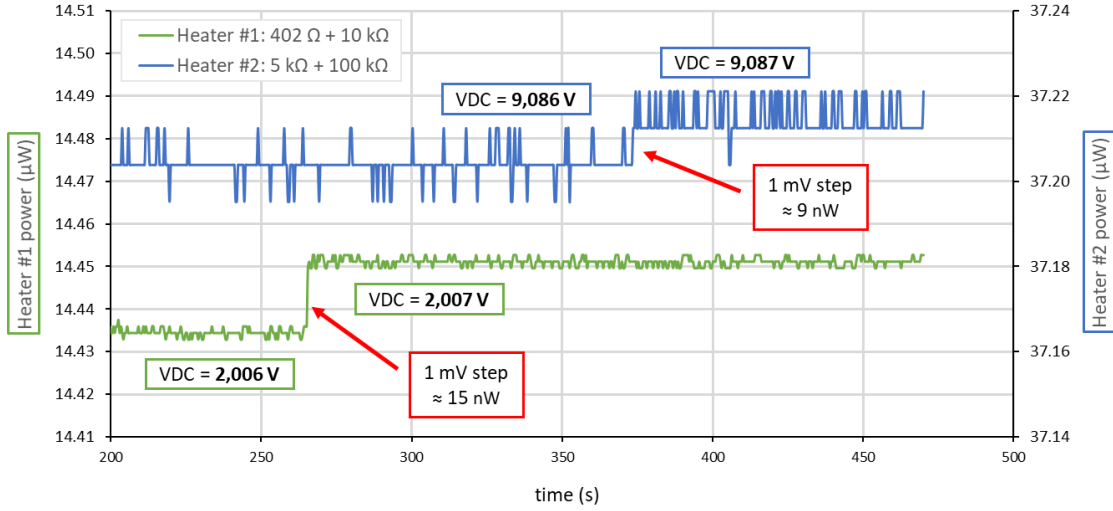


Figure 2.31 – Power measurement precision: addition of 10 k Ω & 100 k Ω external resistances

We note a considerable reduction in the amplitude of the power steps: a 1 mV supply voltage step now leads to a corresponding 15 nW and 9 nW electrical power step for R_1 and R_2 respectively. We also note that for R_2 , the power step is now of the same order as the command and readout noise. This means that the addition of an even larger external resistance would not improve power measurement precision.

2.8 Interfacing

Communication takes place through GPIB or USB interfacing and SCPI commands, all of which are centralised on a LabVIEW program developed for this thesis. Commands are sent to the test rack that themselves command or carry out data readouts on sensors and actuators. The LabVIEW program has 2 crucial roles: time synchronising all measured quantities and ensuring sufficiently fast data acquisition.

2.8.1 Time synchronisation

The LabVIEW program synchronises the time data from multiple sources (e.g. thermometers and accelerometers). An internal time reference ensures identical t_0 times for the different measurement equipment. At each iteration of the code (i.e. a given time step), a single value of each quantity is acquired.

This ensures sufficiently good time synchronisation, precise to within the code run rate. LabVIEW is not a tool for accurate time synchronisation (e.g. to within 1 ms), though this thesis has no such needs.

2.8.2 Data acquisition rate

When a high sample rate is needed (e.g. acceleration measurements), the dedicated data acquisition system is used. Indeed the characteristic time from a mechanical stand point is of the order of a few milliseconds, whereas from a thermal stand point it is of the order of the minute. The code run rate of around of 1 Hz therefore suffices for measurements where high sample rate is not needed.

2.9 Conclusion

To conclude this chapter on the test bench and detection chain, it is worth summarising the range and resolution we are capable of measuring for each quantity, given the capabilities of the described equipment at our disposal. These are summarised in Table 2.24. We note that these working ranges indicate ballpark figures of range and resolution of each quantity of interest we are able to observe.

Table 2.24 – Test bench capabilities: ranges and resolutions of quantities of interest

<i>Name</i>	<i>Range</i>	<i>Resolution</i>
<i>Acceleration</i>	100 μg to 10 mg (10 Hz to 500 Hz)	10 μg (0.1 Hz)
<i>Power dissipation</i>	100 nW to 10 μW	10 nW
<i>Temperature</i>	1 K to 10 K	200 μK
<i>Displacement</i>	100 μm to 1 cm	1 μm

We also note from the discussions of this chapter that as we are working with cryogenics, the test environment is complicated. As we approach absolute zero, the thermal susceptibility of our test bench increases. This allows to observe smaller levels of power dissipation due to microvibrations, though also makes our test bench more sensitive to other heat loads that we do not wish to see. Furthermore the cryostat was not designed for mechanical excitation. This adds another layer of complexity in using it as a microvibration dissipation test bench. For these reasons, the environment must thoroughly be characterised. Together with the measurement methods, these aspects will be presented in chapter 3.

Environment and measurement methods

The previous chapter covered the test bench and detection chain. In this chapter we will present our environment, measurement methods and the interactions between the environment and the test object. This is necessary given the complexity of the mechanical and thermal behaviours we observed.

3.1 Environment characterisation

In this section we present the characteristics of our measurement environment, namely the cryostats. We used 2 cryostats for this thesis. These are the so-called Herschel cryostat and Tedi cryostat, the latter being the smaller one. Both function similarly from a thermal perspective, having identical cooling procedures. The Herschel cryostat – being the larger of the 2 – takes longer to cool but stays cold for longer, making it suited for longer test campaigns. The Tedi cryostat meanwhile is suited for shorter campaigns requiring more versatility, say to adapt the setup between cold phases.

3.1.1 Cryostat mechanical response 10-500 Hz

The purpose of this section is not to identify the cryostat resonances. It is rather to outline our method of ensuring that we can achieve our target acceleration levels at the cold plate ($100 \mu\text{g}$ to 10 mg), across our target frequency band (10 Hz to 500 Hz). We can then confirm that a given cryostat (our measurement environment) is suited to our needs.

In Figure 3.1 we show the mechanical shaker mounted to the exterior of the Herschel cryostat (left) and 3 cryogenic accelerometers mounted orthogonally to each other onto the cold plate (right).

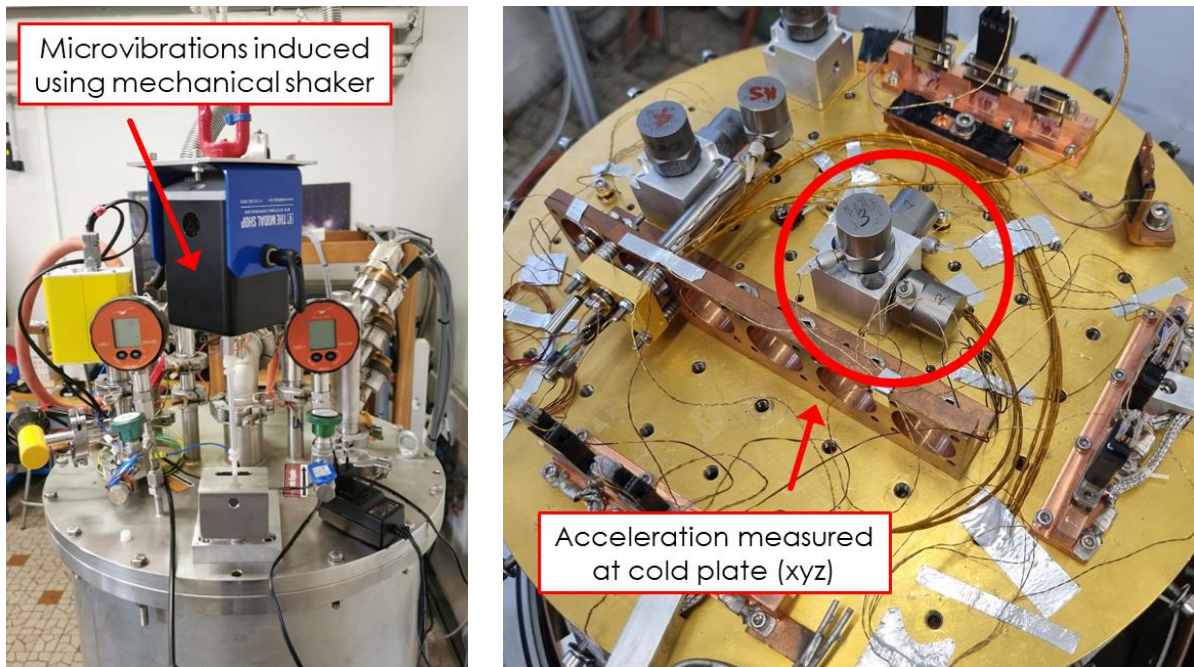


Figure 3.1 – Shaker mounted onto Herschel cryostat exterior (left) and 3 cryogenic accelerometers (xyz) onto cold plate

This setup is depicted in Figure 3.2, showing the vertical excitation of the mechanical shaker (F_z) and the measurement of the components of the acceleration vector at the cold plate (A_x, A_y, A_z).

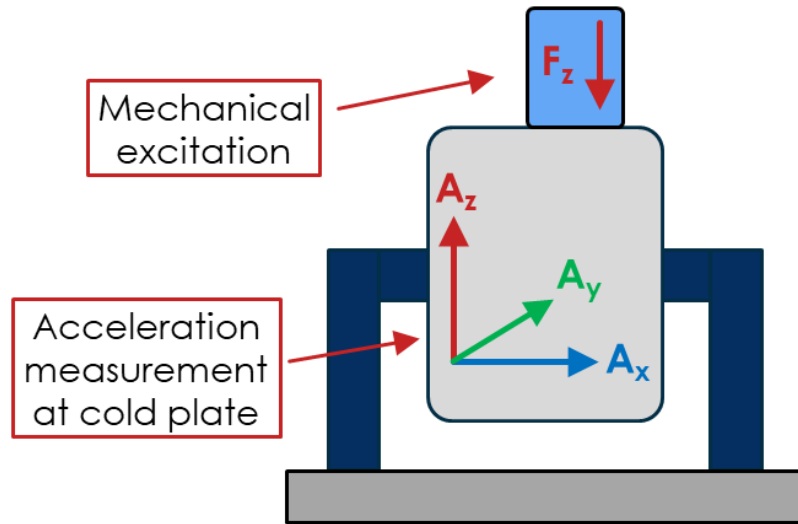


Figure 3.2 – Cryostat mechanical excitation and acceleration measurement at cold plate

Though the mechanical signatures of the 2 cryostats differ, we will only present the results of the Tedi cryostat here. To measure the response we perform sweeps from 10 Hz to 500 Hz, for shaker drive voltages of 250 mV_{pp} and 500 mV_{pp}. The acceleration magnitude (xyz) is shown in Figure 3.3, at 4.2 K. Indeed, when we first characterised the cryostat, we studied the norm of the 3-axis acceleration vector. This gave a first indication of the mechanical transmission from the shaker interface through to the cold plate. As we shall see in this chapter, we later also defined the so-called dissipative acceleration level.

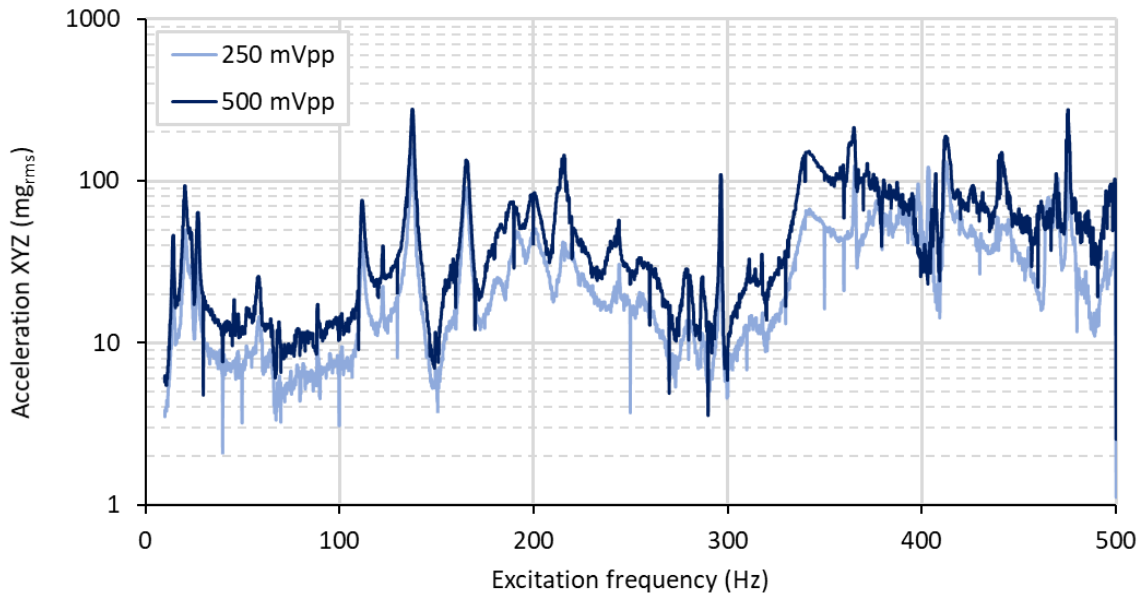


Figure 3.3 – Mechanical response (xyz) of Tedi cryostat at fixed shaker drive voltages at 4.2 K

For this experiment, we used fixed shaker drive voltages, which should result in relatively constant acceleration levels at the interface between the shaker and cryostat, with only slight variations due to the reaction of the cryostat. Despite this, we note many peaks and troughs in the acceleration level at the cold plate. This is expected for any mechanically complex system. For the shaker drive voltages used here, in the 10-500 Hz band, excitation levels are spread across ~2 orders of magnitude (~1 mg_{rms} to ~100 mg_{rms}).

When transmission was very low, we simply increased the shaker drive voltage and found that we never needed to exceed the specified limits of the shaker to reach the target range of 100 μg to 10 mg. Cryostat transmission was therefore never too low.

When transmission was very high, the minimum drive voltage of the function generator (10 mV_{pp}) was sometimes too high to achieve low (sub-mg) acceleration levels. For these cases, we built a 10X voltage divider that we placed between the function generator and the shaker. We could then apply a lower drive voltage and achieve the low target acceleration levels, for these high cryostat transmission cases.

In addition to similar tests carried out with the Herschel cryostat, all in all we concluded that for our working frequency band, we are able to induce the target acceleration levels at the cold plate.

3.1.2 Influence of thermal state on mechanical behaviour

The mechanical transmission to the cold plate was impacted by the thermal state of the cryostat, despite performing identical frequency sweeps. Figure 3.4 shows the acceleration magnitude (xyz) for sweeps from 10 Hz to 50 Hz at several cold plate temperatures, for the Herschel cryostat.

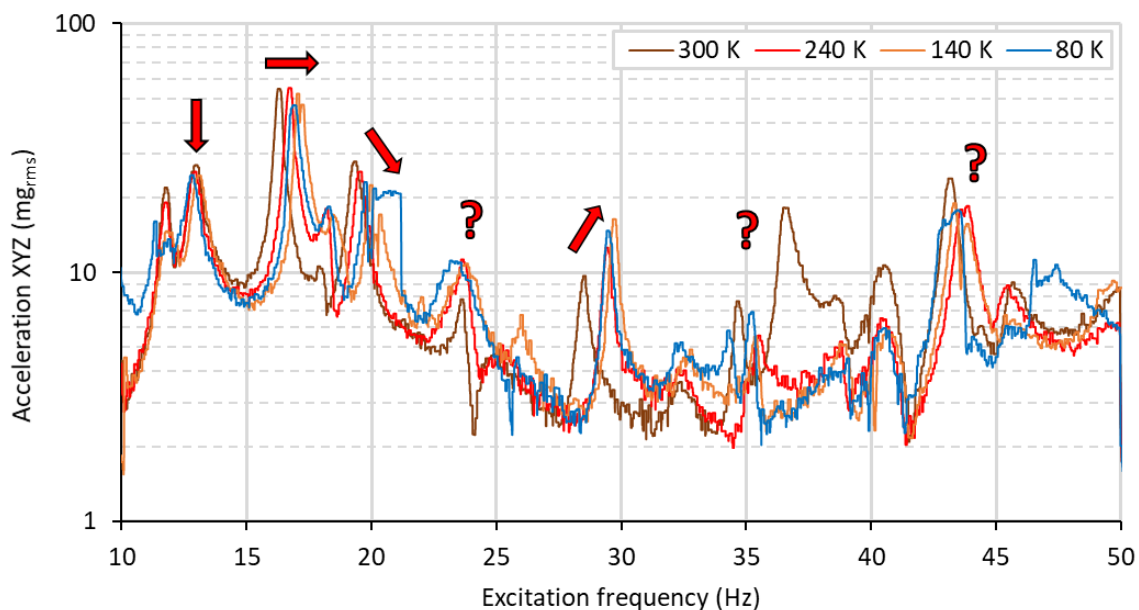


Figure 3.4 – Influence of cryostat state on mechanical response (xyz) of Herschel cryostat at 140 mV_{pp} drive voltage, with arrows indicating frequency shifts and “?” indicating more complex evolution

We note an increase of some resonance frequencies with decreasing temperature. This is expected as the Young modulus (which defines the mechanical stiffness here) increases with decreasing temperature for most materials. For other resonances however, their evolution is erratic.

Regarding the resonance amplitudes, they too are unpredictable: as temperature decreases, some fall whereas others rise. Changes in relative stiffness between multiple parts of the cryostat could in fact compensate each other, leading to frequency shifts and amplitude changes that are difficult to predict. The evolution of the mechanical response of the cryostat with temperature is therefore unpredictable.

We hypothesise that in addition to being influenced by temperature, the cryostat mechanical response also varies due to changes in cryogen mass. Thermal stabilisation times may also play a role, in particular for large and well insulated parts of the cryostat. We now specifically study the impact of cryogen mass.

3.1.3 Influence of cryogen mass on mechanical behaviour

Figure 3.5 shows additional acceleration measurements, taken for varying quantities of cryogen in the reservoirs and at varying temperatures. We note the generally large variations in the mechanical response, despite carrying out identical sweeps from 150 Hz to 220 Hz at a drive voltage of 500 mV_{pp}.

For the two measurements at 77 K (left side of the figure), with the nitrogen reservoir empty (green) and full (orange), the response varies a lot despite being measured at similar temperatures. We note a fivefold change in peak acceleration. This suggests the nitrogen mass damps the mechanical response. Indeed when the reservoir is full (the Herschel cryostat has a 60 L nitrogen capacity as per Table 2.5) the nitrogen mass at 77 K is ~48 kg. This is considerable and likely explains the very different mechanical responses.

For the 2 measurements at 300 K (right side of the figure) that were carried out 2 hours apart (solid black and dashed black) and after warm up, the internal temperature of the cryostat was still evolving. This emphasizes that the cold plate temperature is not sufficient to predict the cryostat behaviour.

Generally, it seems that a combination of higher temperature and reduced cryogen mass increases mechanical transmission between the shaker and cold plate in our frequency range of interest.

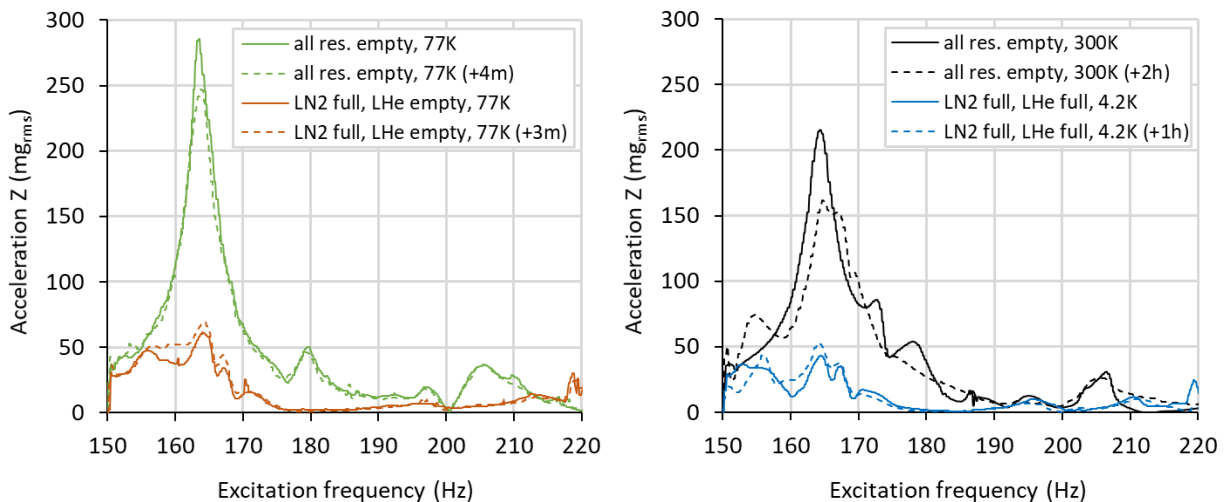


Figure 3.5 – Influence of nitrogen mass (left), temperature and cryogen mass (right), on mechanical response (z) of Tedi cryostat at 500 mV_{pp} drive voltage

This complex response of the cryostat – influenced by cryogen mass and temperature (which is only measured at the cold plate) – implies that its mechanical behaviour can change significantly with time.

3.1.4 Cryostat mechanical response at 1.2 K and 80 K

Finally, our experimental conditions at 1.2 K over the 10-50 Hz range is shown in Figure 3.6, together with the response at 80 K shown previously. There is a notable difference between the responses, which

cannot be explained by a change in the Young modulus or thermal contraction as these mostly occur between 300 K and 100 K. The discrepancy is thus likely due to mechanical influence of the helium mass.

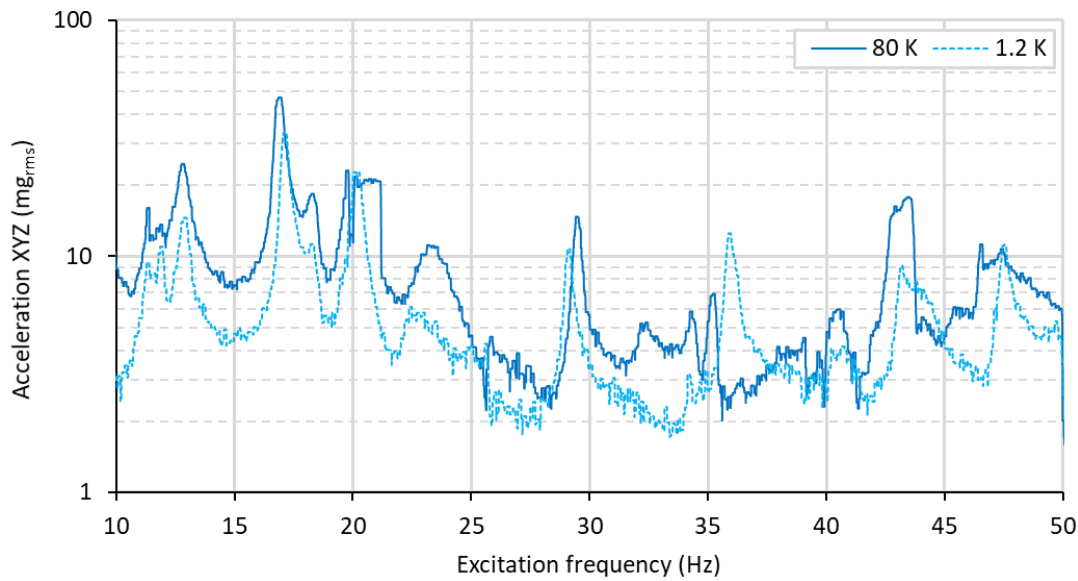


Figure 3.6 – Influence of helium mass on mechanical response (xyz) of Herschel cryostat at 140 mVpp drive voltage

Indeed, at 1.2 K the helium reservoir contains superfluid helium-4. However, when cooling from 4.2 K to 1.2 K by pumping the bath (discussed in section 2.5.3), roughly half the helium is consumed. Starting with a full reservoir at 4.2 K (the Herschel cryostat has a 32 L helium capacity as per Table 2.5), we expect to have ~16 L of superfluid helium-4 at 1.2 K. This corresponds to a helium mass at 1.2 K of ~2.3 kg.

3.1.5 Temperature stability

The temperature stability of our cold plate directly affects the quality of our microvibration dissipation measurements. We characterized it in two configurations: helium bath at atmospheric pressure (4.2 K) and pumped bath at 1.2 K. The temperature stability for both cases is shown in Figure 3.7.

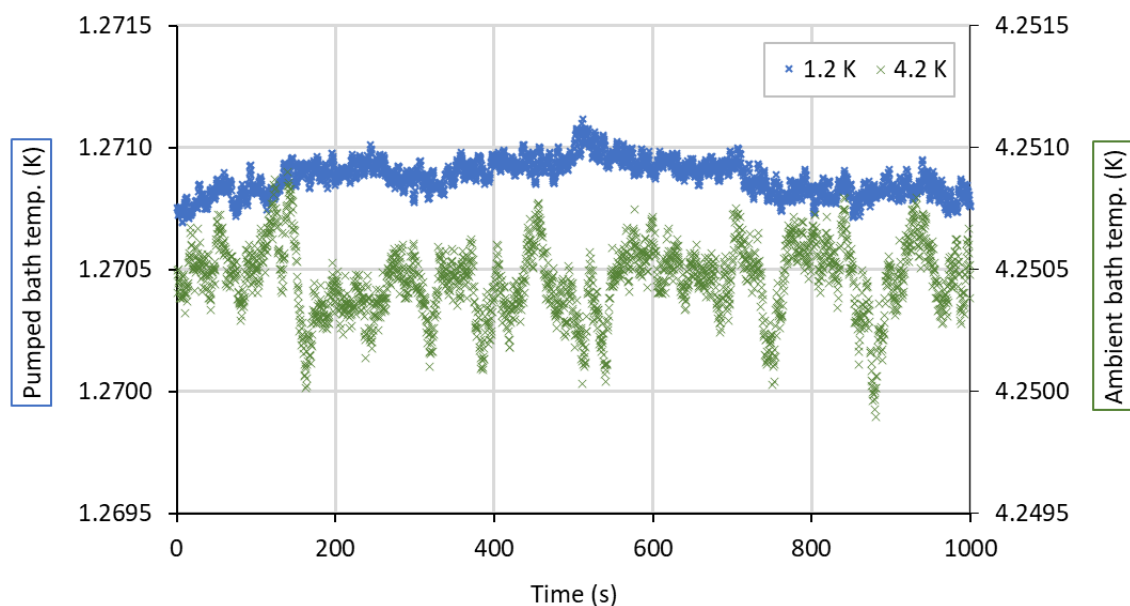


Figure 3.7 – Cold plate temperature stability at 4.2 K and 1.2 K measured using thermometer #3 (X84246) and CABTR

The full temperature scale is just 2 mK. The cold plate temperatures at 4.2 K (green) and 1.2 K (blue) are measured over a duration of 1000 s. Thermometer #3 (X84246) was used for these measurements (see Table 2.17). We find that pumping significantly improves the stability of our cold plate temperature.

We can quantify the stability of these cold plate temperature measurements, once again using confidence levels in terms of the number of standard deviations σ (i.e. the coverage factor). For a coverage factor of $k=2$ (95% confidence), the temperature stabilities at 4.2 K and 1.2 K are given in Table 3.1.

Table 3.1 – Cold plate temperature stability at 4.2 K and 1.2 K

4.2 K	1.2 K
296 μ K	140 μ K

The cold plate temperature is about twice as stable at 1.2 K as at 4.2 K. We also find that the values are greater than the temperature *readout* noise levels deduced in section 2.7.1.2 (see Table 2.17), which is expected. For thermometer #3 (X84246), these were 140 μ K (at 4.2 K) and 21 μ K (at 1.2 K).

The measured temperature fluctuations are partially due to the detection chain (CABTR) noise and the thermometer's characteristics, but should mostly reflect true bath temperature fluctuations.

We conclude that it is preferable to work in the pumped bath configuration due to temperature stability. Most experiments were carried out with the cryostat in the pumped bath configuration at 1.2 K.

3.2 Other environment characteristics

3.2.1 Change in translational acceleration direction

For the first months of this thesis, acceleration was only measured in the direction of excitation of the shaker, namely the vertical axis (z). Indeed, it was assumed that the acceleration level at the cold plate due to microvibrations was predominantly vertical if the shaker was mounted in this way. It was also assumed that the orientation in 3-dimensional space of the acceleration vector was constant, irrespective of the excitation frequency. We found however that these 2 assumptions were incorrect.

Here we study the direction of the acceleration vector at the cold plate. We first present broadband sweeps (10-500 Hz), followed by narrowband sweeps (200-230 Hz) at the critical frequencies of our test object.

3.2.1.1 Broadband behaviour

Figure 3.8 shows a 10-500 Hz frequency sweep carried out at a shaker drive voltage of 500 mV_{pp} on the Tedi cryostat. The translational acceleration level at the cold plate, along the cryostat pivot axis (x), the 2nd horizontal axis (y) and the vertical axis (z) were measured simultaneously. The accelerometers were mounted to the cold plate as close as possible to the interface with our test object.

We note that the z component is generally larger than the x and y components (as expected), though the horizontal components are not as negligible as previously thought. At certain excitation frequencies they are of the same order as the vertical component, at for example 20 Hz, 110 Hz and 240 Hz.

These changes in the relative amplitudes mean that the acceleration vector is rotating in 3-dimensional space as the sweep progresses. Measuring it solely along the direction of excitation is therefore not fully representative of the microvibration level that causes dissipation (it is underestimated).

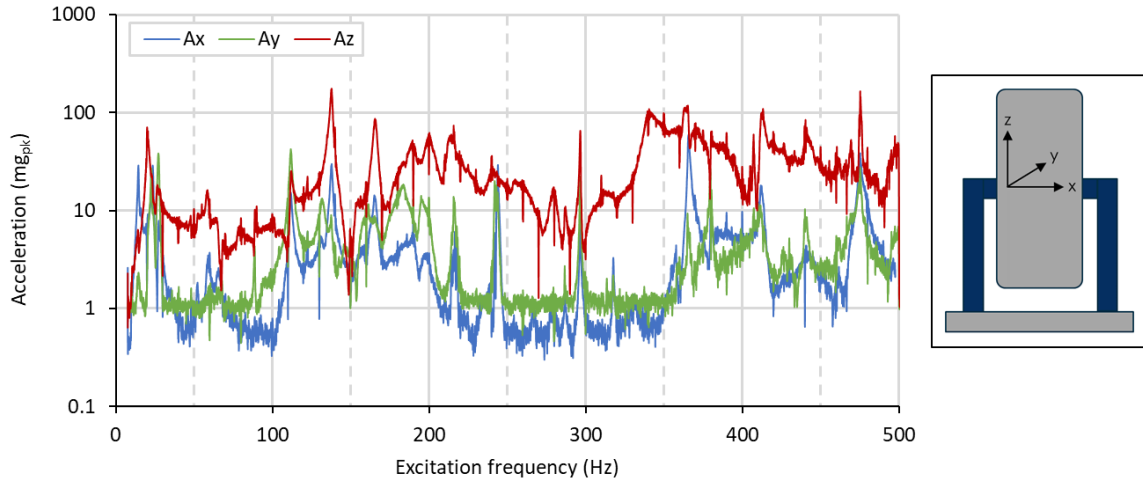


Figure 3.8 – Translational acceleration components at cold plate of Tedi cryostat, frequency sweep at 500 mV_{pp} drive voltage and z-axis shaker excitation, at 4.2 K

3.2.1.2 Narrowband behaviour

Next, we also wished to study the behaviour of the acceleration vector for small bandwidths around the resonance of our test object (the heat switch). This resonance was first found at ~213 Hz at 1.2 K.

The component along the heat switch axis (*y*) excites a higher mode than the 1st cantilever mode, whose microvibration dissipation behaviour is not studied. We therefore measured the translational components in the transverse plane: *A_z* (vertical) and *A_x* (horizontal). Figure 3.9 shows them measured in 2 cases.

In the left side of the figure, we excited at 300 K at fixed frequencies from 212.8 Hz to 213.4 Hz with a 0.1 Hz interval. The right side shows tests at 1.2 K, at 213.1 Hz (where microvibration dissipation was later observed) and at an arbitrary frequency of 200 Hz. For all tests, we used 3 shaker drive voltages.

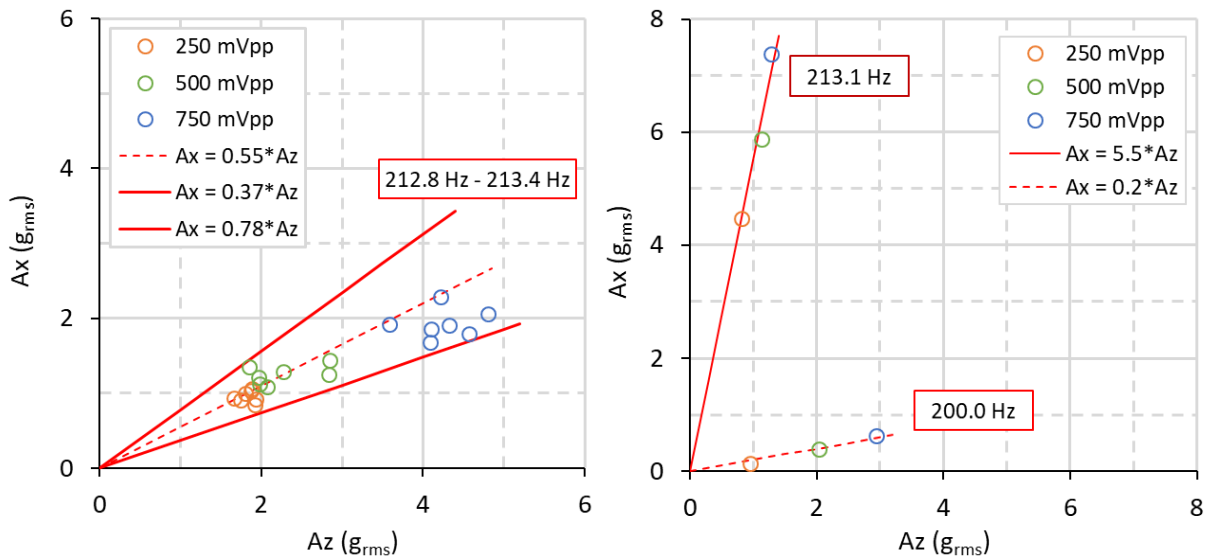


Figure 3.9 – Translational acceleration components at cold plate of Herschel cryostat, for fixed frequency excitations at 300 K (left) and 1.2 K (right)

From the left side, we observe that as the shaker drive voltage increases, the spread of data points does too. This could indicate mechanical instability at higher excitation levels. As represented by the 2 solid red lines, the relative amplitude of the acceleration components also varies a lot: the amplitude of A_x varies between roughly 37% and 78% of the amplitude of A_z . This means the acceleration vector at the cold plate is changing direction as the excitation frequency progresses, even over a narrow 0.6 Hz band.

On average the relative amplitude of A_x is about 55% that of A_z here, represented by the dotted red line. However, in this case the norm of the transverse acceleration A_{xz} is only 14% greater than A_z . This is sufficiently low for the analysis of the preliminary measurements presented in chapter 4, where we only mounted 1 accelerometer. The error is later addressed by measuring several acceleration components.

From the right side of the figure, we note an even larger shift in the relative amplitudes for excitation frequencies further apart. At 213.1 Hz, the horizontal component is now over 5 times larger than the vertical component. This could be explained by the presence of the test object resonance at this frequency at 1.2 K. At 200 Hz, the horizontal component is just 20% of the vertical one. These relative amplitudes correspond to an acceleration vector direction in the transverse plane of $\sim 80^\circ$ and $\sim 11^\circ$ respectively from the horizontal direction. Between the excitation frequencies that are just 13 Hz apart, we observe a large rotation of the acceleration vector. It is therefore crucial that we measure the acceleration level in multiple directions and not only the direction of excitation.

3.2.2 Presence of rotational acceleration

In addition to the translational acceleration (A_x, A_y, A_z), we also studied the rotational acceleration around the heat switch axis R_y . This is not to be confused with the change in direction of the translational acceleration vector we have just seen. To study R_y , we mounted 2 accelerometers vertically (A_{z1} and A_{z2}) on both sides of the heat switch and swept from 200 Hz to 230 Hz (thus around the heat switch resonance) at a drive voltage of 800 mV_{pp}. Figure 3.10 shows peak acceleration plotted against excitation frequency.

The mode at ~ 217 Hz, where peak amplitudes are equal, could indicate either pure translation or pure rotation. We cannot differentiate between these modes as phase information is lost in this figure. The difference in amplitudes at ~ 205 Hz suggests that a rotation axis exists that is not centred between the accelerometers, causing the peak amplitude of one accelerometer to be lower than the other.

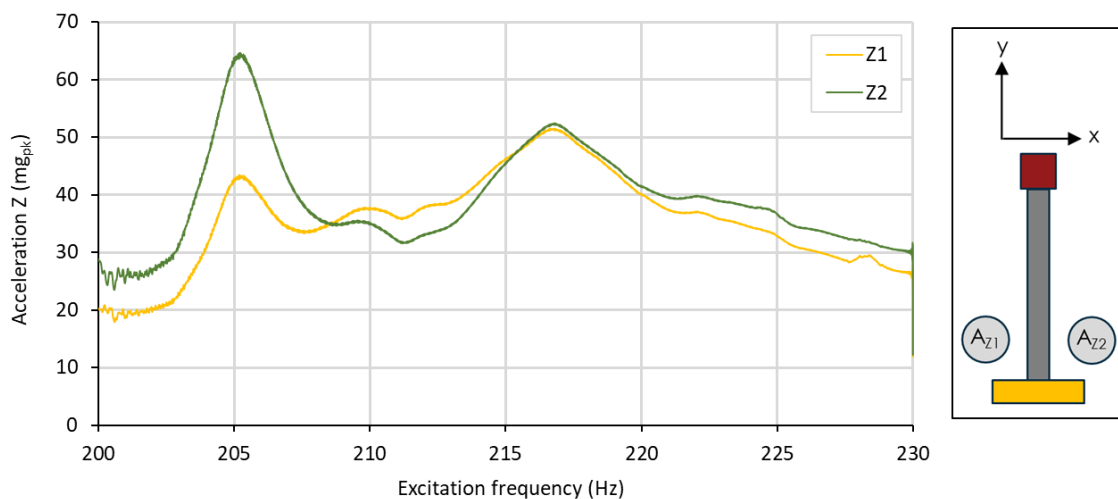


Figure 3.10 – Vertical acceleration component at cold plate of Tedi cryostat, frequency sweep at 800 mV_{pp} drive voltage

We then plot the 2 vertical components against each other, shown in Figure 3.11. For pure translation we expect $A_{Z2} = A_{Z1}$ (perfectly in phase) and for pure rotation $A_{Z2} = -A_{Z1}$ (perfectly out of phase). We also add a colour scale for the excitation frequency. We restrict our plot here to excitation from 200 Hz to 220 Hz.

We note that for higher excitation frequencies (red) the cold plate is mostly translating and barely rotating. This could indicate the 1st drum mode of the circular cold plate. At lower excitation frequencies (blue), we observe a narrow ovaloid shape, still indicating the presence of translation but now also slight rotation.

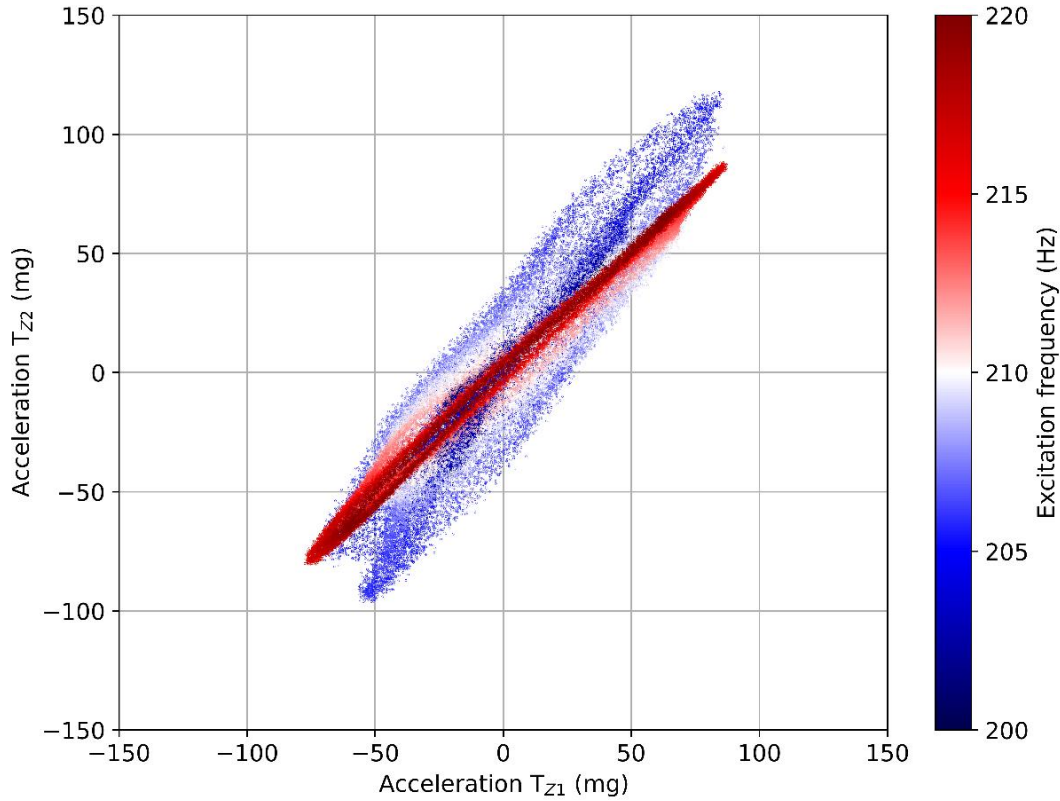


Figure 3.11 – Lissajous plot of acceleration components to observe the potential rotation of the cold plate of Tedi cryostat, frequency sweep at 800 mV_{pp} drive voltage

3.3 Summary of environment

We now summarise the past 2 sections, that have presented the mechanical and thermal characterisation of the environment (section 3.1), and other characteristics we believe were worth presenting (section 3.2).

We characterized our cryostat and showed non-linearity of its mechanical response to the shaker excitation. Moreover, we found this mechanical response to vary greatly with experimental conditions: temperature and cryogen mass. These observations need to be taken into account for our measurements.

We also find that the acceleration components that are not in the shaker excitation direction are not always as negligible as initially thought. The 1st cantilever mode of the heat switch that we wish to study may be excited by both transverse directions. We should therefore consider the norm of the acceleration vector in the transverse plane as it is more representative of the dissipative acceleration. The variations in the mechanical response of the cryostat also justifies the need for synchronous acceleration and dissipation measurements, such that the true level of mechanical excitation is known for a given dissipation level.

Lastly, we conclude here that we are able to achieve the acceleration levels desired (100 μg to 10 mg), despite the amplitude and direction of the acceleration vector varying a lot with the excitation frequency. Next, we will present our methods for dissipation measurement.

3.4 Dissipation measurement methods

Dissipation is at the core of my work and to measure it two methods were employed. This section will present them as well as the methods used to induce microvibrations.

We measure dissipation in an insulated object in two ways. The first is from the measurement of temperature elevations, due to the heat dissipation of microvibrations. The second is using temperature regulation. A drop in the regulation power of the object directly corresponds to microvibration dissipation.

The method used for inducing microvibrations has an impact on the method used for measuring their dissipation. For inducing microvibrations, we impose a fixed shaker drive voltage at either a fixed frequency (when observing temperature elevations) or with a frequency sweep (when using temperature regulation). The combinations of methods that we are able to use together will now be presented.

3.4.1 Temperature slope at fixed frequency

3.4.1.1 Method

Fixed frequency excitation allows to study the response, both mechanical and thermal, of the test object at a specific frequency and shaker drive voltage. In this thesis, when inducing microvibrations at fixed frequencies we measure dissipation through the measurement of temperature elevation.

The 1st mode resonance frequency of our test object was measured at 300 K by means of the laser sensor. When cooled, we set the desired excitation level and frequency through the function generator. The signal output to the shaker is then switched on for a defined duration (generally 60 s), before being switched off. The process is repeated for each excitation level and frequency we wish to study.

At specific fixed frequencies near the resonance of our test object, temperature elevations due to microvibrations were observed on our insulated test object.

The rising temperature slopes of the vibration phases are then calibrated to the rising temperature slopes of known electrical heating levels at the same temperature. From this we obtain dissipation for each discrete excitation frequency, at a given fixed shaker drive voltage and temperature. Figure 3.12 shows a theoretical example using the generalised thermal model presented in chapter 1.

Discrete dissipation levels (2 μW , 5 μW and 10 μW) are applied to the insulated mass, simulating microvibration dissipation levels at fixed frequencies. They each last 60 s, with 10 min rest between them to allow the temperature to fall back down. We impose a fixed thermal resistance of $7.87 \times 10^4 \text{ K/W}$ and capacitance of $6.74 \times 10^{-3} \text{ J/K}$, which roughly correspond to the characteristics of our test object at 4 K.

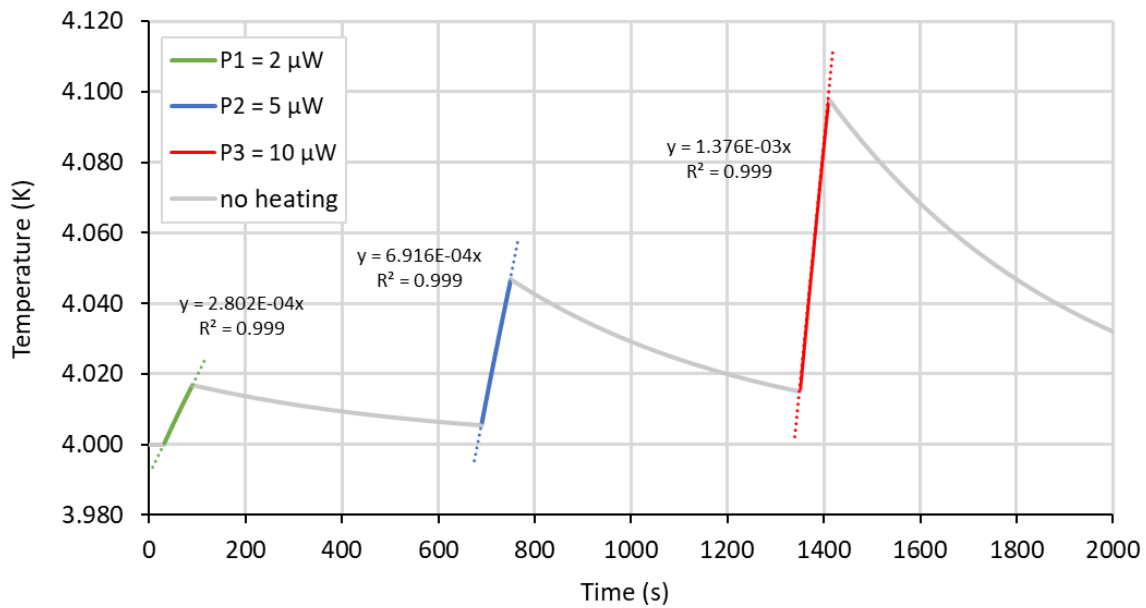


Figure 3.12 – Calculated temperature elevations of insulated mass for imposed dissipation lasting 60 s

The generalised thermal model states that the initial temperature slope is given by P/c . We thus deduce the dissipation level graphically from the slopes of the temperature elevations labelled in Figure 3.12 and the capacitance. The dissipation levels deduced in this way are listed in Table 3.2.

Table 3.2 – Dissipation levels deduced through calculated temperature elevation slopes

$P_{imposed}$	Slope	$P_{calculated}$	Error
2 μW	2.80×10^{-4} K/s	1.89 μW	-5.5%
5 μW	6.92×10^{-4} K/s	4.66 μW	-6.8%
10 μW	1.38×10^{-4} K/s	9.30 μW	-7.0%

If the capacitance is not known for a given temperature, we employ the calibration step discussed above using electrical heating. We observe that there is a small underestimation of the dissipation level. This is one of several limitations of the temperature slope method. These will now be discussed.

3.4.1.2 Limitations

A main limitation of the temperature slope method is the error between the measured and true dissipation level seen in Table 3.2. This is likely due to an underestimation of the initial slopes given the curvature of the temperature elevations. As the linear fits are made over a 60 s heating duration, the true initial slope is indeed slightly higher. This is more evident in Figure 3.13, where for the same system parameters we impose 10 μW for 1000 s (yellow). We now underestimate the initial slope P/c (green) by 58%.

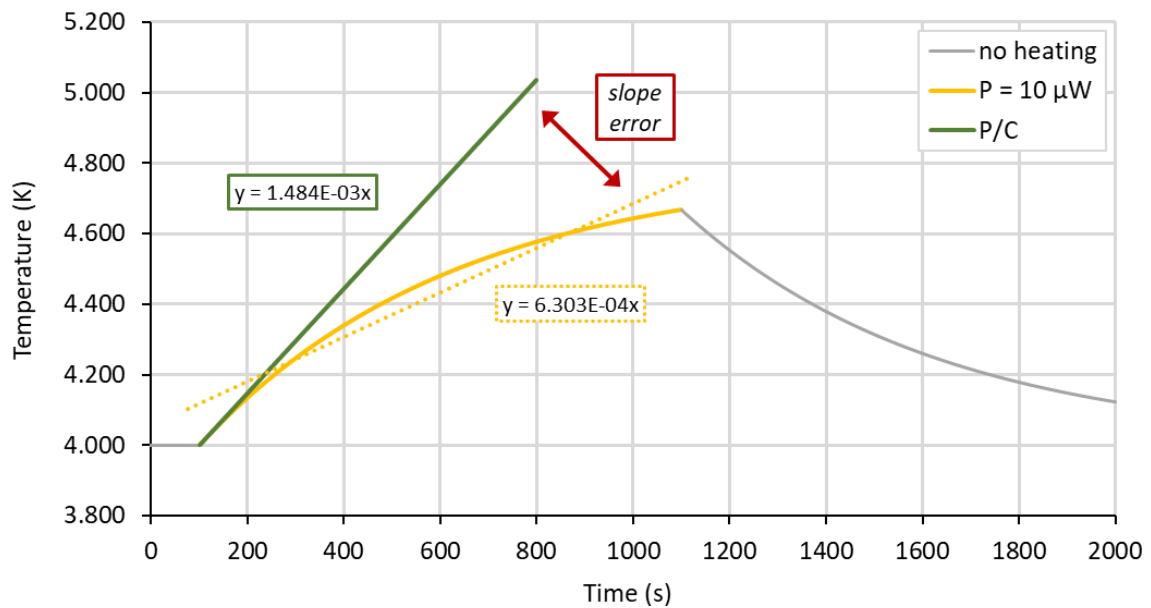


Figure 3.13 – Calculated temperature elevation (yellow) of insulated mass for 10 μW lasting 1000 s and initial slope (green). Values given to illustrate example.

The heating durations should therefore be kept short for a more precise estimation of the initial slope. When they last 60 s the error is less than 10%, which is not problematic for the interpretation of our experimental results, as we are interested in the orders of magnitude of microvibration dissipation.

A second limitation of this method is that calibration is only valid at a given temperature. Due to the changes in thermal conductivities and heat capacities, temperature slopes say for 500 nW of dissipation at 4.2 K and 500 nW at 1.2 K, are not equal. We must thus repeat calibration at each working temperature.

Another drawback of this method is that measurements induce temperatures changes, even if only slightly. This changes material properties and so heat fluxes between parts, other than microvibration dissipation, that are not accounted for. Dissipation is then measured less accurately. To avoid this, we would have to wait for the temperature slopes to return to zero (i.e. infinitely long) before every measurement or calibration step. At best, we impose long rest intervals between heating phases.

A last drawback of the temperature slope method is that it is easier to implement in a non-continuous way. For broad dissipation curves (low Q factors) this is not too limiting, as discrete frequencies predict dissipation relatively well. For sharp dissipation curves (high Q factors) however, the frequency step between points must be small enough to ensure dissipation is properly resolved, especially near the peak. More data points at discrete frequencies lengthens the already tedious (versus frequency sweeps) measurement and calibrations steps.

3.4.2 Temperature regulation for frequency sweeps

3.4.2.1 Linear or logarithmic sweep

To measure microvibration dissipation continually, we use temperature regulation in conjunction with frequency sweeps. The motivation for using frequency sweeps rather than discrete fixed frequencies is to obtain the continuous response, both mechanical and thermal, of the test object over a frequency band.

For given start and end frequencies, sweeps can be linear or logarithmic. When linear, the sweep speed (in s/Hz) is constant. As much time is spent sampling a given frequency width. When logarithmic, the sweep speed grows exponentially for ascending frequency. Expressed in units of octaves/min, as much time is spent sweeping from 1 Hz to 10 Hz as from 10 Hz to 100 Hz. In this way, an equal amount of periods of oscillation are measured for each frequency (10 Hz for 10 s is as many periods as 100 Hz for 1 s). A logarithmic sweep is useful when excitation frequency spans several decades (e.g. 1-1000 Hz).

However, in this thesis we mostly study narrow excitation bands, having say 10 Hz bandwidths (or even less). This is especially true for experiments where dissipation is also measured. This is because the thermal time constants involved are always much larger than the mechanical time constants, meaning sweep speeds must be very low (as slow as 1 hour to sweep 1 Hz) during this thesis. This can lead to very long duration tests, if we sweep across too high bandwidths. Thus, given the generally narrow frequency bands of this thesis, linear frequency sweeps are preferred over logarithmic ones.

3.4.2.2 Temperature control

In this method, temperature regulation is achieved through PID control (whose parameters also depend on the chosen sweep speed). We regulate the temperature of an insulated mass using a heater and choose the set point such that the regulation power is larger than the expected microvibration level. If underestimated, regulation power falls to zero and the temperature rises.

We also avoid excessively high regulation temperatures as it implies higher voltage and current ranges, which potentially increases power noise. High regulation temperatures also induce unwanted increases in thermal conductance and capacitance. As seen in the thermal model of chapter 1, this reduces our sensitivity to heat loads (e.g. microvibration dissipation). We discuss this further in chapter 4.

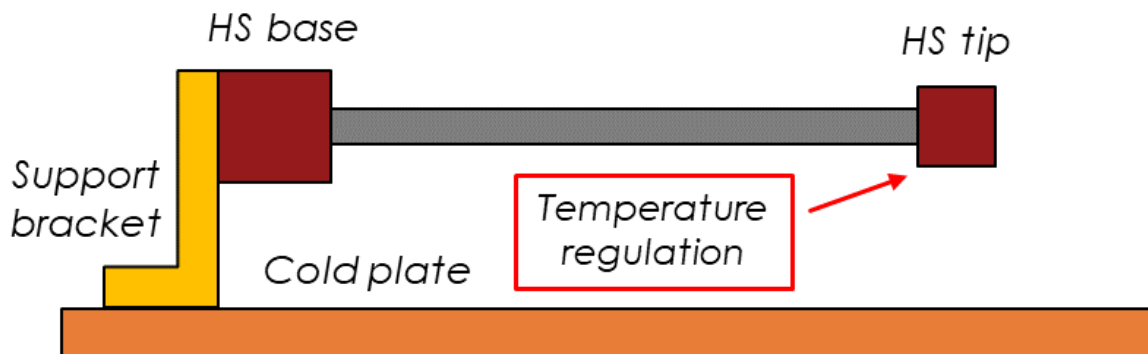


Figure 3.14 - Temperature regulation of heat switch tip

Figure 3.15 shows an example of temperature regulation of the tip of a heat switch. For a set point of 2 K at the tip of HS2 (red) and a cold plate temperature (orange) close to 1.25 K, the regulation power (blue) settles close to 800 nW. Having already identified the dissipative frequency window of 219-221 Hz, we observe that a frequency sweep across this range causes a drop in the regulation power of about 550 nW. Zeroing the power regulation at its pre-microvibration level and taking the mirror image, we directly obtain microvibration dissipation arriving at the tip as a function of excitation frequency, for a given drive voltage.

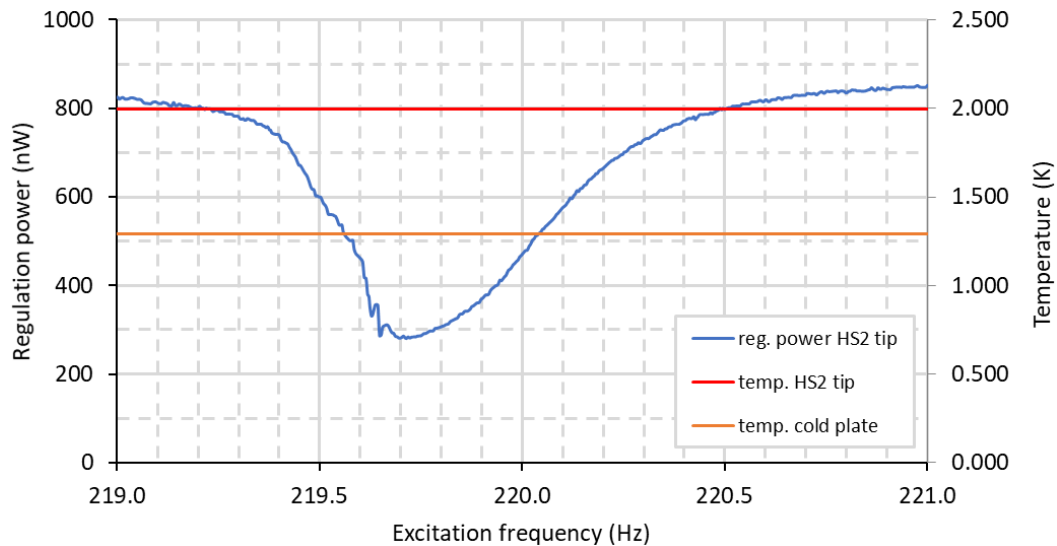


Figure 3.15 – Temperature regulation example with drop of regulation power due to microvibration dissipation in HS2

Unlike for the temperature slope method, temperature regulation can be used on individual insulated components of the same test object. This is because temperatures remain stable and so heat fluxes between the components too. In this way the heat flux arriving at a given location due to microvibration dissipation can be differentiated from heat flux due to another insulated component at another regulation temperature. For example, for a specific configuration of the heat switch, we regulated the temperatures of both the base and tip. Dissipation occurring at the tip did not affect the regulation power at the base (and vice-versa), as the total heating at a given location (dissipation plus regulation power) remained constant. We could therefore measure dissipation simultaneously at each location.

Temperature regulation also avoids the need for calibration at several dissipation levels and each working temperature. It also provides a continuous dissipation curve as a function of excitation frequency, as opposed to discrete dissipation points at fixed frequencies. The method nevertheless has its drawbacks.

3.4.2.3 Limitations

We studied the impact of the sweep speed and direction (ascending or descending) on measured dissipation. In Figure 3.16 we show dissipation curves of the tip of HS2 for different speeds and directions.

Each point on the regulation power curve should represent a thermal equilibrium at the corresponding excitation frequency, such that the set point temperature is maintained and the temperature slope is zero. If the sweep speed is too high however (say 100 s/Hz), we spend less time exciting at dissipative frequencies. The peak dissipation level therefore appears lower than when exciting at a lower speed (say 500 s/Hz). Hence if dissipation occurs over a narrow frequency window (as in Figure 3.15), we may underestimate its amplitude if the sweep speed is too high. This is one of the main drawbacks of this measurement method.

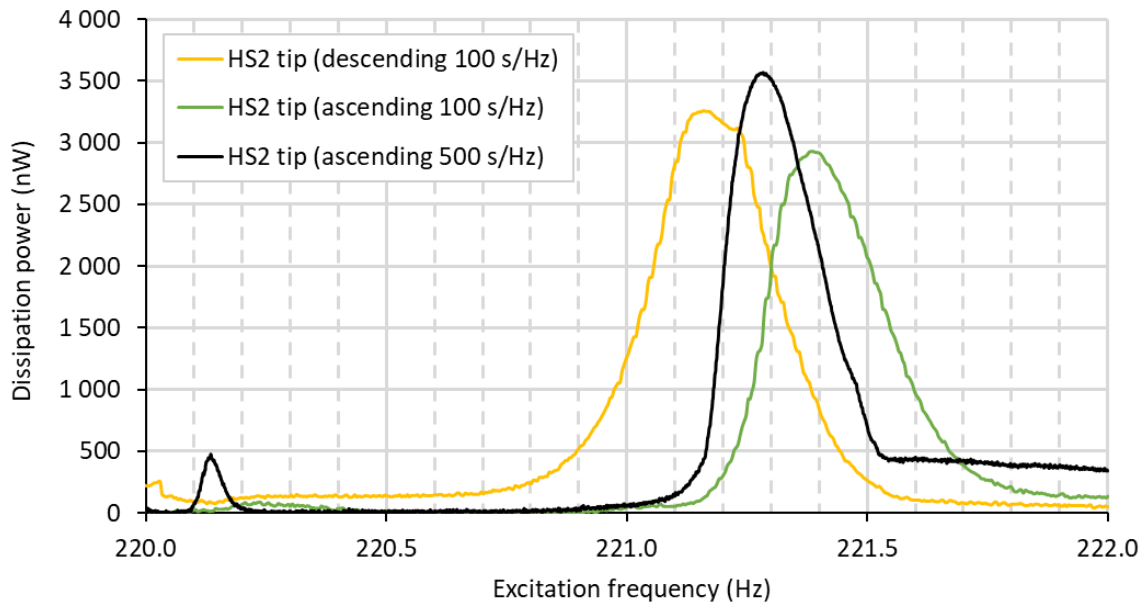


Figure 3.16 – Impact of sweep speed and direction on microvibration dissipation curves

A second drawback concerns the peak dissipation frequency. At 100 s/Hz, the peaks for the descending (yellow) and ascending (green) tests do not appear at the same excitation frequency: ~221.15 Hz versus ~221.4 Hz (an error of ~125 mHz from the intermediate value for a sweep speed of 100 s/Hz).

For a lower speed of 500 s/Hz (black), peak dissipation lies between the first 2 peaks. We do not expect the peak dissipation level and frequency to depend on the sweep speed or direction. It is therefore a measurement error rather than a phenomenological observation. Again it can be explained by too high sweep speeds: thermal equilibrium is not perfectly maintained. Ideally, the sweep speed should be infinitely slow for the test object to continually be in thermal equilibrium.

We deduce that in this case, a sweep speed of 500 s/Hz is slow enough, as its dissipation peak falls roughly half way between the 2 peaks at 100 s/Hz. In general, we adapt the sweep speed depending on the test object and resonance width. We will indicate the chosen speed for each test.

Another drawback of temperature regulation, is that the PID control needed for it introduces noise in the regulation power. It is in particular the P (proportional) control that is responsible for this, as any noise present in the temperature readings is simply amplified by a factor P. We can remedy this by manually reducing the P control when the temperature is close to the set point. Regulation power noise is then reduced, but so too is PID reactivity, potentially amplifying the effects visible on Figure 3.16. A set of PID parameters and chosen sweep speed are deemed adequate when the dissipation curves from ascending and descending frequency sweeps are similar (i.e. peak dissipation amplitudes and frequencies overlap).

3.5 Test object interaction with cryostat

We suspect that the resonance of our test object impacts the acceleration level at the interface between it and the cryostat. This section will present the mechanical interaction observed between the 2 and how this may affect the quality of our microvibration dissipation measurements.

3.5.1 Mechanical feedback of test object

Acceleration peaks at the cold plate appear at similar frequencies as the microvibration dissipation peaks of our test object (discussed in chapter 4). These acceleration peaks are visible in Figure 3.17 for different test configurations that showed different Q factors. Here we show the magnitude of the acceleration in the transverse plane (yz) of our heat switch test object (that is along x).

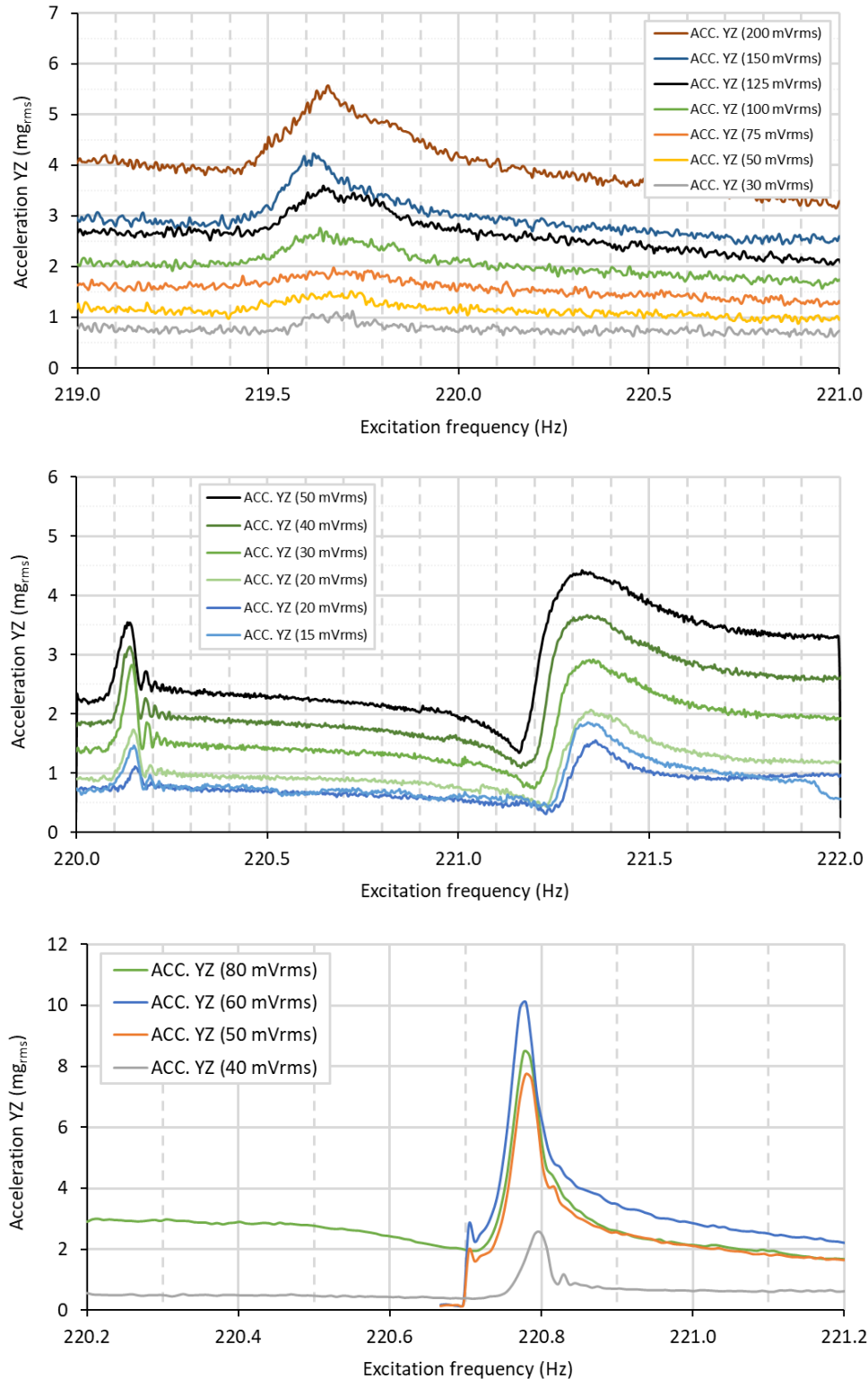


Figure 3.17 – Injected acceleration (yz) levels at the cold plate due to mechanical feedback of heat switch for different test configurations that showed different Q factors

We theorise that there is mechanical feedback from the test object to the cold plate, as depicted in Figure 3.18. We did not expect this given the much lower mass of our test object compared to the cryostat.

However, for an oscillating mass of say 10 g and a Q factor of 1000, the test object will produce the equivalent force of a rigid mass of 10 kg at a given acceleration level. The resonance of a small test object having a high Q value could thus impact the acceleration level at the cold plate.

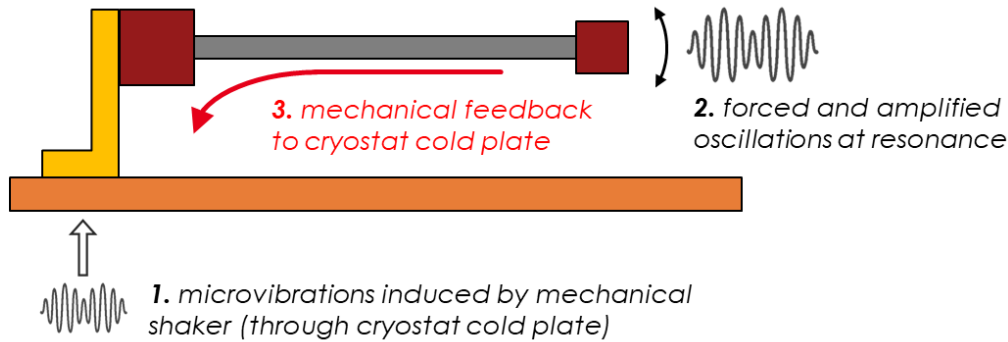


Figure 3.18 – Heat switch mechanical feedback due to amplification of oscillations at resonance

We also studied the impact of the resonance frequency of a heat switch on the displacement of the other. We found mechanical interaction between the 2 heat switches mounted onto the same bracket, and so choose to mount only 1 heat switch at a time when possible. This is discussed further in Appendix 2.

3.5.2 Precision in Q factor calculation

We also wished to study whether the precision with which we measured the Q factors was impacted by the shape of the injected acceleration profile. Through the model presented in chapter 1, we can predict the output acceleration level (that is to say at the heat switch tip) theoretically, by imposing an injected acceleration level, Q factor and resonance frequency. If we can obtain the output acceleration profile, the Q factor can be deduced in 2 ways: either from the mechanical amplification at resonance or using the bandwidth definition of the Q factor. As will be described in chapter 4, at cryogenic temperatures we can only deduce the Q factor using the bandwidth definition, as only the dissipation curves are at our disposal (and not the output acceleration profile).

Here we shall therefore study the change in the measured Q factor when using the bandwidth definition, being the method used with the dissipation curves. As the study case, we take the injected acceleration level of the highest drive voltage of the second configuration of Figure 3.17 (middle). It is shown again in Figure 3.19 in black, together with an arbitrary flat profile set at its peak of 4.4 mg_{rms} in red.

We also impose a Q factor of 1500 and choose to study 3 frequencies: 220.5 Hz, 221.2 Hz and 221.3 Hz, as the exact mechanical resonance frequency at cryogenic temperatures is not known for most of our tests. The frequencies are highlighted by red dots in Figure 3.19. They are chosen such that they correspond, from left to right, to a flat region, steep region and peak of the measured acceleration profile.

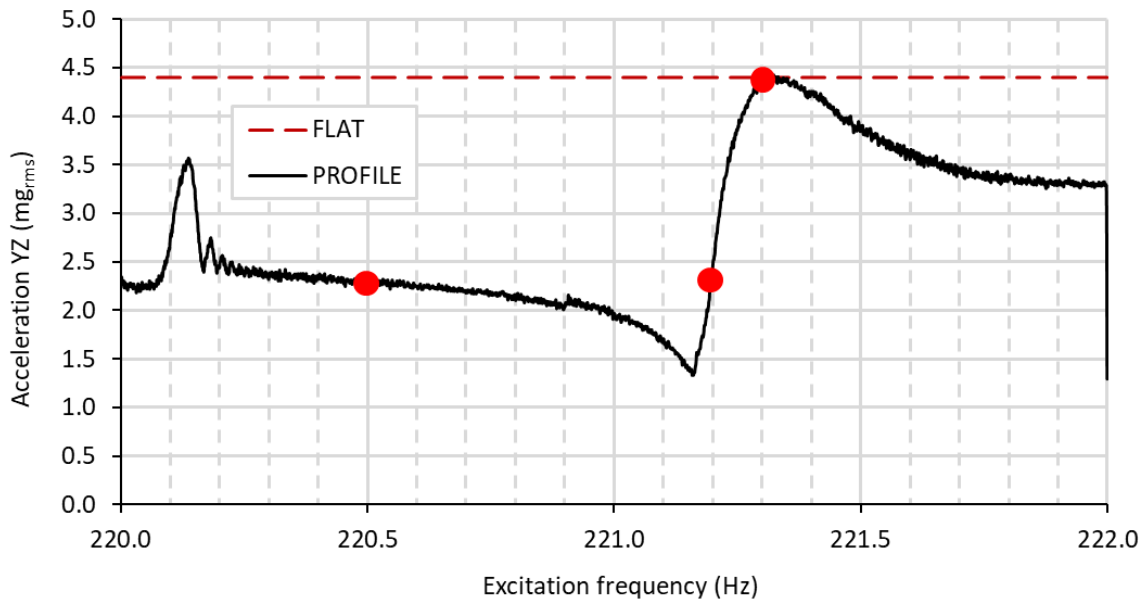


Figure 3.19 – Injected acceleration profiles: measured in 2nd test configuration (black) and arbitrary flat profile at 4.4 mg_{rms}

The resulting output acceleration profiles, as predicted by the model for a heat switch resonance set to 221.3 Hz, are shown in Figure 3.20. We observe perfect symmetry for the flat case (red), whereas when the measured acceleration profile is considered the output acceleration profile (black) is distorted. This consequently introduces an error when deducing the Q factor using the bandwidth definition.

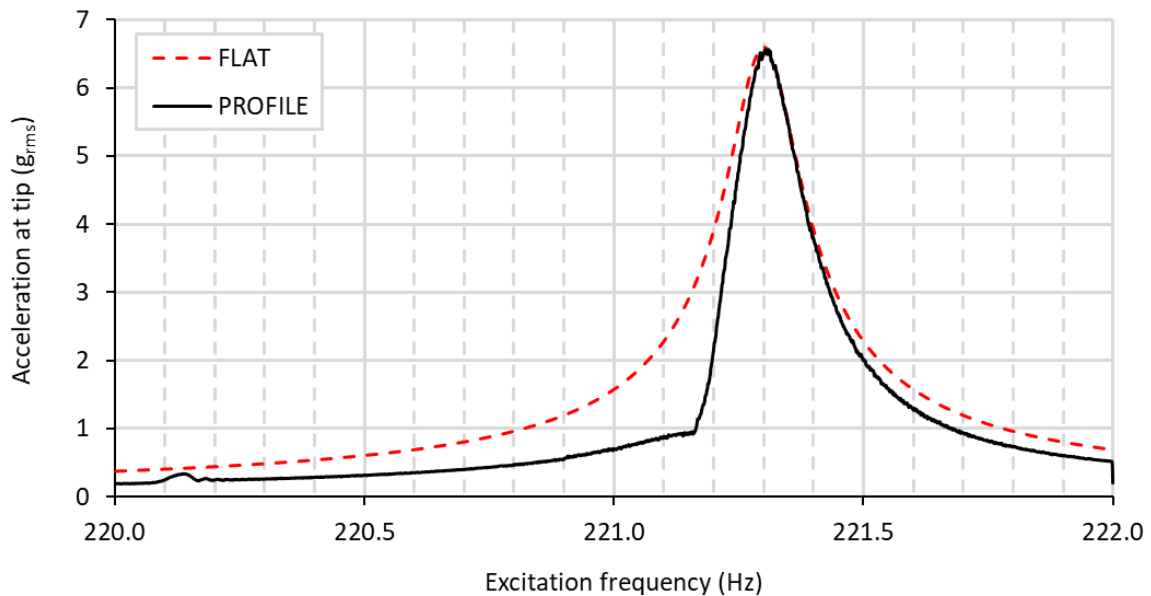


Figure 3.20 – Output acceleration profiles according to mass-spring model

Table 3.3 shows the calculated Q factor values when imposing the flat acceleration profile or the measured acceleration profile, for the 3 resonance frequencies. When assuming a flat acceleration level, the deduced Q factor is very close to the imposed value of 1500 (as expected). The slight discrepancy is due to the discretisation of the model. When accounting for the non-flat acceleration profile in our model, we observed varying degrees of error in the Q factor calculation. It is highest (18.9%) when the acceleration profile is steepest and lowest (2.3%) when it is most flat, which could be expected.

Table 3.3 – Error in Q calculation due to non-flat injected acceleration profile

f_d	Q_{flat}	$Q_{profile}$	ΔQ
220.5 Hz	1494	1528	2.3%
221.2 Hz	1501	1849	18.9%
221.3 Hz	1499	1766	15.1%

When calculating the Q factor from the dissipation curves, we in effect assume the injected acceleration profile is flat. However for our tests, as we have seen, the true acceleration level at the cold plate is in fact not flat due mechanical feedback from our test object (as well as variations in the cryostat's transmission, as seen previously). From Table 3.3 we conclude that the Q factor calculation error does not exceed 20% when the Q factor is not too high and so neither are the mechanical feedback and dissipation distortion.

For a specific test configuration however, we noted particularly high Q factors: the injected acceleration profiles shown in the bottom of Figure 3.17 caused corresponding dissipation curves that were highly distorted. This introduced large errors in the calculation of the Q factor from the bandwidth definition. For such cases (very high Q factor), we must therefore be cautious in the interpretation of Q factors calculated from dissipation curves.

3.5.3 Summary

Mechanical feedback from the test object to the cold plate is expected in proportion to the mass of the test object and the Q factor of its resonance. As long as the Q factors of our test object are not too high, this effect should not dramatically impact our measurements and analysis, particularly the calculation of the Q factor, but care will have to be taken for larger Q factor values or masses of the test object.

3.6 Conclusion

This chapter first discussed the characterisation of our measurement environment, a helium bath cryostat. Depending on the state of the cryostat (a combination of temperature, cryogen mass and thermal stabilisation times), we found large variations in the mechanical response at the cold plate (~2 orders of magnitude). The direction of the translational acceleration changed depending on the excitation frequency and we noted the presence of slight rotational acceleration about the heat switch axis.

Using a mechanical actuator that functions in cryogenic conditions, mounted onto the cold plate near the heat switch interface could flatten the response. For our tests however, no such actuator was available.

We also presented mechanical feedback observed from our main test object (the heat switch) to our measurement environment (the cryostat). This would not occur were we to use a mechanically ideal test setup, say where the test object is mounted rigidly onto a high-mass interface, such as the vibration shaker that is briefly used for mechanical characterisation (discussed in chapter 4). This too would allow us to inject a flat acceleration profile to the heat switch attached to it

Hence for future tests incorporating dissipation measurements, we must be cautious of the sensitivity of the mechanical response at the cold plate to the state of our cryostat and to our test object.

In this chapter we also presented the 2 dissipation measurement methods (temperature slopes or temperature regulation) and the 2 ways in which we induce microvibrations (fixed frequency or frequency sweeps). In the following chapter we will present the wide range of observations made possible with them.

Tests with gas-gap heat switch

We may now present the microvibration dissipation measurements of this thesis. We will start with preliminary measurements on which we will base the refinement of our measurements. The remainder of the chapter then presents evolutions of the test configuration, which allowed us to carry out our most complete and advanced test campaigns, and that produced the most significant results of this thesis.

4.1 Preliminary dissipation measurements

The preliminary dissipation measurements were carried out with an elementary setup and were a good way for us to validate our experimental approach on a simplified test bench. The main goal of this section is to show the first observation of microvibration dissipation of this thesis, and present typical orders of magnitude that we are able to measure, given the capabilities of our test bench seen in chapter 2.

4.1.1 Rigid base configuration

As we progress through this chapter, several heat switch test bench configurations will be presented. The simplified setup enabled preliminary dissipation measurements. They were the first evidence of microvibration dissipation seen in this thesis and will be presented in this section.

In the so-called rigid-base configuration, the base of the heat switch is rigidly fixed to a support bracket, by means of four M3 screws. The heat switch is shown schematically in this configuration in Figure 4.1.

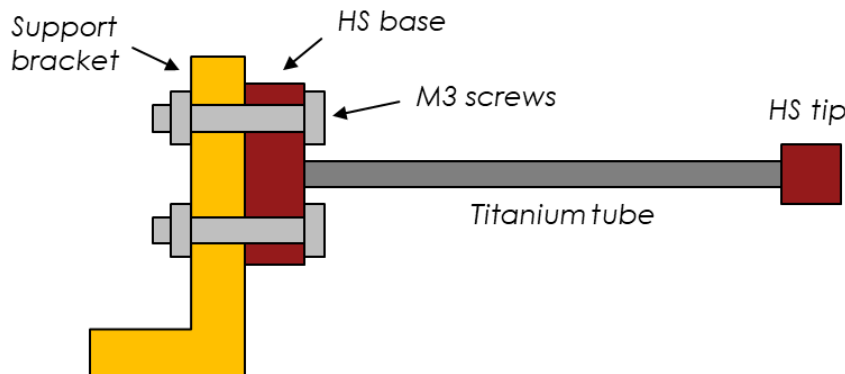


Figure 4.1 – Heat switch mounted with M3 screws (rigid base configuration)

The base and bracket both made of copper, together they can be considered a single element, both thermally (homogeneous and equal in temperature) and mechanically (perfect mechanical transmission between them). Regarding instrumentation, a thermometer and electrical heater (see chapter 2) are affixed to the heat switch tips. Figure 4.2 shows HS1 and HS2 mounted onto the cold plate.

In the left side of Figure 4.2, in the rear, we see a displacement limiter mounted onto HS1. This so-called snubber protects the heat switch from excessively high vibrations, particularly during launch. It is mounted to the base, concentrically to the titanium tube and tip of the heat switch. There is a 300 μm radial gap between them. As the vibration levels induced during this thesis are not even close to those observed during launch, the snubber has no real use here and does not affect the results presented. It nevertheless allows us to differentiate visually between HS1 (with snubber) and HS2 (without snubber).

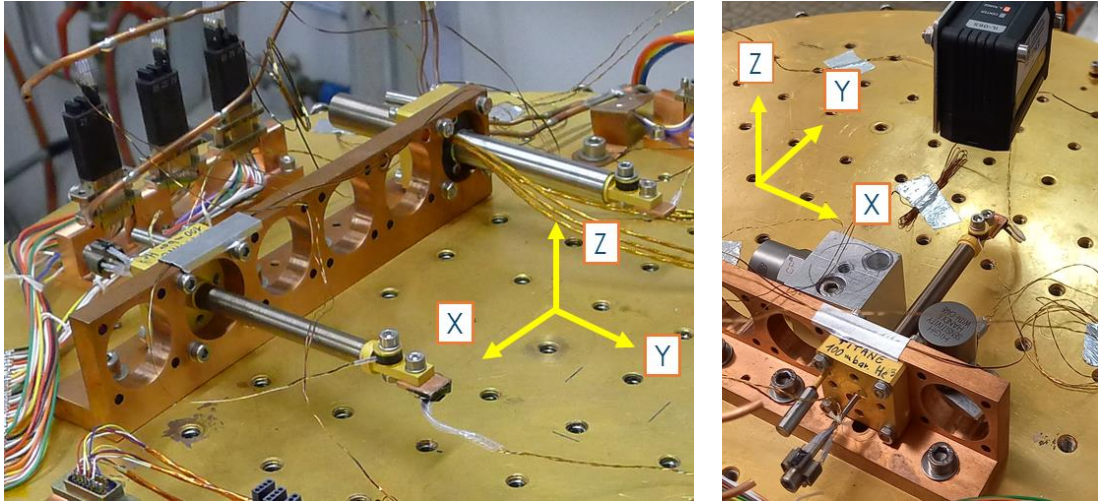


Figure 4.2 – HS1 (left, rear, with snubber) and HS2 (left, front, without snubber) mounted in rigid base configuration. Laser sensor pointing onto HS2 tip and 2 cryogenic accelerometers mounted onto cold plate and orthogonally to each other (right).

The right side of Figure 4.2 shows HS2, the laser sensor pointed at its tip and 2 cryogenic accelerometers (described in chapter 2) mounted onto the cold plate. They are orthogonal to each other and in the transverse plane of the heat switch axis. For the preliminary dissipation measurements presented in this section however, we only measured the vertical component (z) of the acceleration.

The first observations of the dissipation due to induced microvibrations can now be presented.

4.1.2 Temperature elevations

A direct measurement of the thermal susceptibility to microvibrations is the effect on temperature. Dissipation of microvibrations could induce temperature elevations. We present the first observations of temperature elevations in our test object here, based on the method described in section 3.4. We then compare temperature elevations observed at 1.2 K and at 4.2 K.

4.1.2.1 First observations

In the rigid base configuration, when cooled to 1.2 K and when inducing microvibrations at various fixed frequencies, temperature elevations were observed at the tip of HS1 near its 1st mode resonance frequency (HS2 was not tested with this method). This is shown in Figure 4.3, where the coloured portions represent vibration phases, while during the grey portions no microvibrations are injected. The vibration phases are of equal duration, lasting 60 s each. The shaker drive voltage is fixed at $V_s = 750 \text{ mV}_{pp}$ here. This does not per se impose a constant acceleration level at the cold plate, due in particular to variations of the cryostat transmission versus excitation frequency (see section 3.5). Despite only 0.1 Hz steps between the different vibration phases, we note different temperature elevations due to microvibration dissipation, the largest being at 213.1 Hz. As this dissipation appears over a very narrow frequency band, this already indicates a large Q factor.

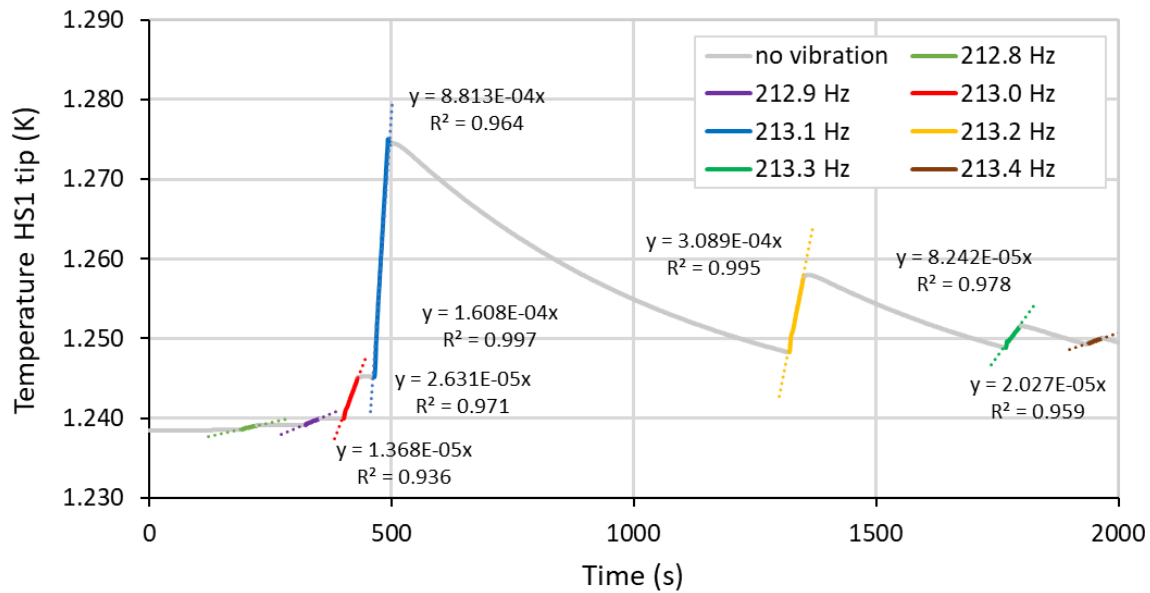


Figure 4.3 – Temperature elevations of the tip of HS1 due to microvibrations at various fixed frequencies at 1.2 K and at fixed shaker drive voltage

4.1.2.2 Comparison between test at 1.2 K and at 4.2 K

We carried out the same microvibration dissipation tests as discussed 4.1.2 that were done at 1.2 K, but now at around 4.2 K. The new temperature elevations observed at the tip of HS1 are shown in Figure 4.4.

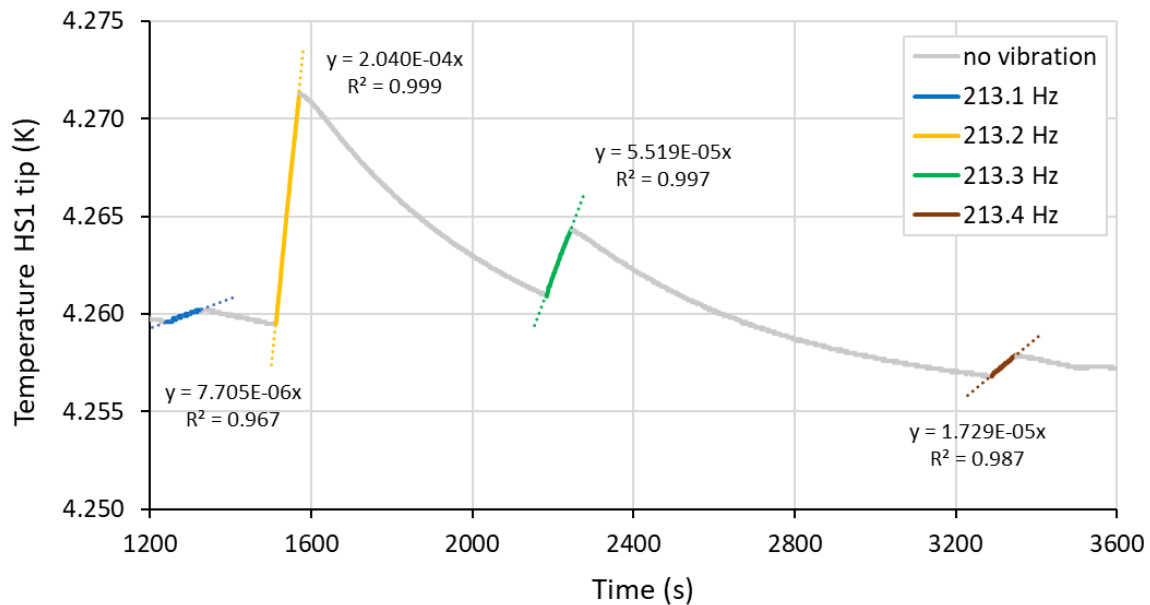


Figure 4.4 – Temperature elevations of HS1 tip due to induced microvibrations at various fixed frequencies at 4.2 K and at fixed shaker drive voltage

The temperature slopes at each frequency and both temperatures can now be compared. These first 2 tests were both at a shaker drive voltage of $V_s = 750 \text{ mV}_{pp}$. They are summarised in Table 4.1.

Table 4.1 – Observed temperature slopes of HS1 tip for $V_s = 750 \text{ mV}_{pp}$

Excitation frequency	1.2 K	4.2 K
212.8 Hz	$1.4 \times 10^{-5} \text{ K/s}$	<i>within noise floor</i>
212.9 Hz	$2.6 \times 10^{-5} \text{ K/s}$	<i>within noise floor</i>
213.0 Hz	$16 \times 10^{-5} \text{ K/s}$	<i>within noise floor</i>
213.1 Hz	$88 \times 10^{-5} \text{ K/s}$	$0.77 \times 10^{-5} \text{ K/s}$
213.2 Hz	$31 \times 10^{-5} \text{ K/s}$	$20 \times 10^{-5} \text{ K/s}$
213.3 Hz	$8.2 \times 10^{-5} \text{ K/s}$	$5.5 \times 10^{-5} \text{ K/s}$
213.4 Hz	$2.0 \times 10^{-5} \text{ K/s}$	$17 \times 10^{-5} \text{ K/s}$

In Table 4.1 we note that the temperature slopes are higher at 1.2 K than at 4.2 K. This represents a gain in sensitivity due to the lower working temperature. We will discuss this in more detail later in this chapter.

At 1.2 K, the same tests were also carried out with 2 lower shaker drive voltages, namely $V_s = 500 \text{ mV}_{pp}$ and $V_s = 250 \text{ mV}_{pp}$. The additional temperature slopes are summarised Table 4.2. Here we observe that the temperature slopes are lower for lower shaker drive voltages, as expected.

Table 4.2 – Observed temperature slopes of HS1 tip at 1.2 K for 2 additional shaker drive voltages

Excitation frequency	$V_s = 500 \text{ mV}_{pp}$	$V_s = 250 \text{ mV}_{pp}$
212.8 Hz	<i>within noise floor</i>	<i>within noise floor</i>
212.9 Hz	<i>within noise floor</i>	<i>within noise floor</i>
213.0 Hz	$3.1 \times 10^{-5} \text{ K/s}$	$4.8 \times 10^{-6} \text{ K/s}$
213.1 Hz	$5.3 \times 10^{-4} \text{ K/s}$	$1.4 \times 10^{-4} \text{ K/s}$
213.2 Hz	$2.5 \times 10^{-4} \text{ K/s}$	$1.2 \times 10^{-4} \text{ K/s}$
213.3 Hz	$4.4 \times 10^{-5} \text{ K/s}$	$2.2 \times 10^{-5} \text{ K/s}$
213.4 Hz	<i>within noise floor</i>	<i>within noise floor</i>

Lastly, in both tables we note that at some discrete frequencies, temperature elevations are so small (or non-existent) that they fall within the test bench temperature noise floor. In all cases, it is only over a very narrow frequency band that temperature elevations due to the dissipation of microvibrations are observed.

4.1.3 Calibration using electrical heating

As discussed in chapter 3, one of the main drawbacks of the temperature slope method is that for calibration, measured temperature elevations are only valid at a given temperature. As calibration is temperature-dependant, it must be done for the elevations at both 1.2 K and 4.2 K. Between vibration

phases, we must also wait for the temperature to drop back down before beginning a new vibration phase. It is interesting to verify that we obtain equal dissipation at both temperatures. We expect the level of microvibration dissipation to be almost independent of the working temperature (at low temperatures).

Now, having evaluated the temperature slopes caused by different levels of microvibrations and at different temperatures, we calibrate these slopes to known levels of electrical heating. The choice of levels of electrical heating used for calibration were such that they caused temperature elevation slopes of the same order as those observed due to microvibration dissipation shown Figure 4.3 and Figure 4.4. They were found to be roughly between 100 nW and 1500 nW. The further the results of dissipation tests move outside of this range, the less reliable the calibration.

In Figure 4.5 fixed levels of electrical heating were applied to the tip of heat switch HS1 for a duration of 60 s, at around 1.2 K and 4.2 K. Shown are the temperatures of the tip of HS1 over time. The coloured portions of the curves represent heating phases, while during the grey portions no heating is applied.

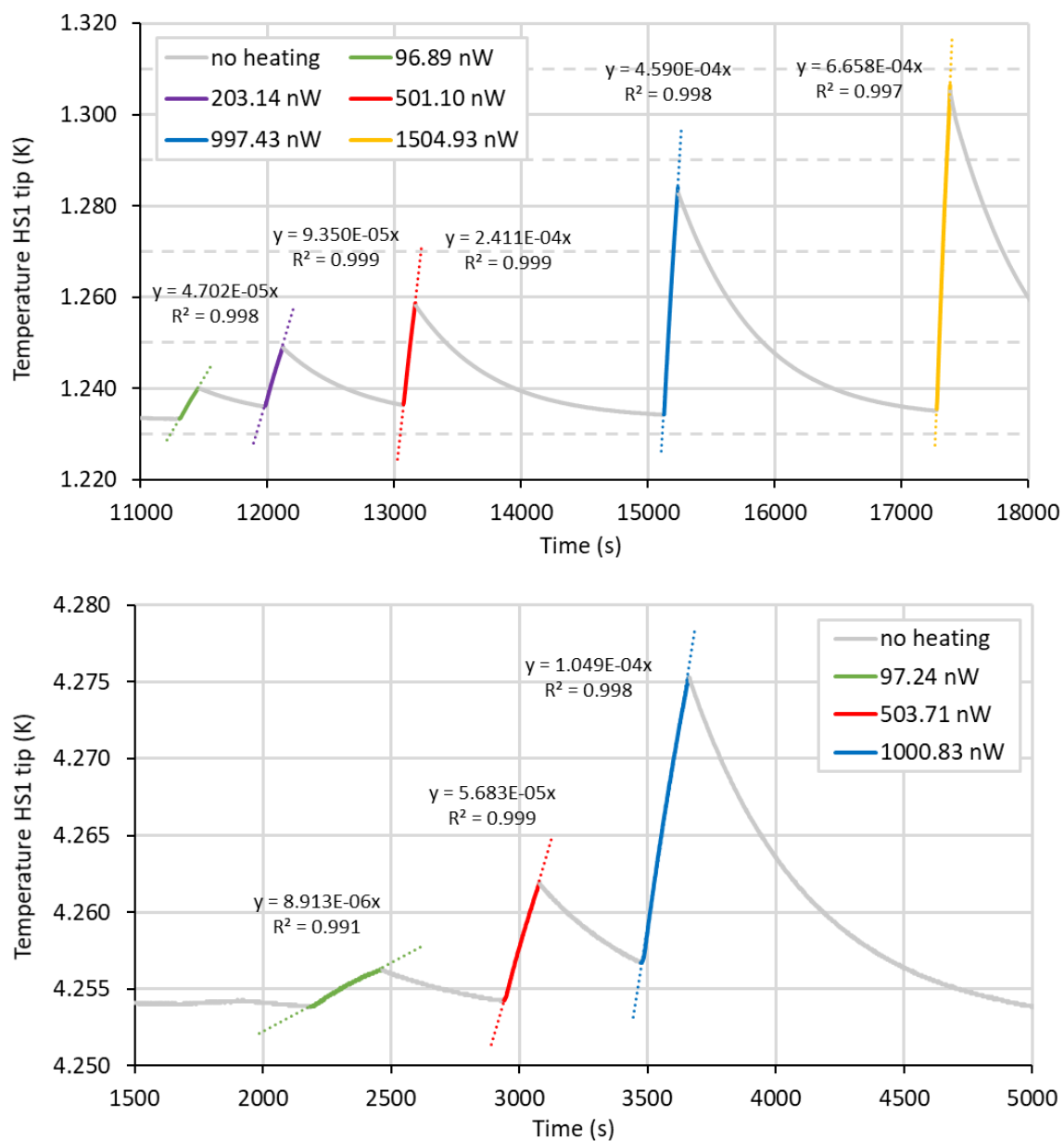


Figure 4.5 – Temperature elevations of HS1 tip due to fixed electrical heating at 1.2 K (top) and 4.2 K (bottom)

We note in Table 4.3 the temperature elevation and calculated average slope for each 60 s heating period.

Table 4.3 – Temperature elevations and calculated slopes of HS1 tip for given electrical heating at 1.2 K and 4.2 K

Electrical power	Temperature elevation		Calculated slope	
	1.2 K	4.2 K	1.2 K	4.2 K
100 nW	~7 mK	~2 mK	4.7×10^{-5} K/s	0.9×10^{-5} K/s
200 nW	~13 mK	-	9.4×10^{-5} K/s	-
500 nW	~22 mK	~8 mK	2.4×10^{-4} K/s	0.6×10^{-4} K/s
1000 nW	~49 mK	~18 mK	4.6×10^{-4} K/s	1.0×10^{-4} K/s
1500 nW	~70 mK	-	6.7×10^{-4} K/s	-

The corresponding calibration curves, giving the calculated temperature slopes for each injected electrical heating level, are shown in Figure 4.6 for 1.2 K (blue) and 4.2 K (green). Linear fits are applied to each and made to pass through the origin. The slopes of these fits (expressed for convenience in mK/s/nW) inform us on the sensitivity of the tip of HS1 to dissipation, in other words its thermal susceptibility to microvibration dissipation. The higher this sensitivity, the more precisely a heat load is resolved.

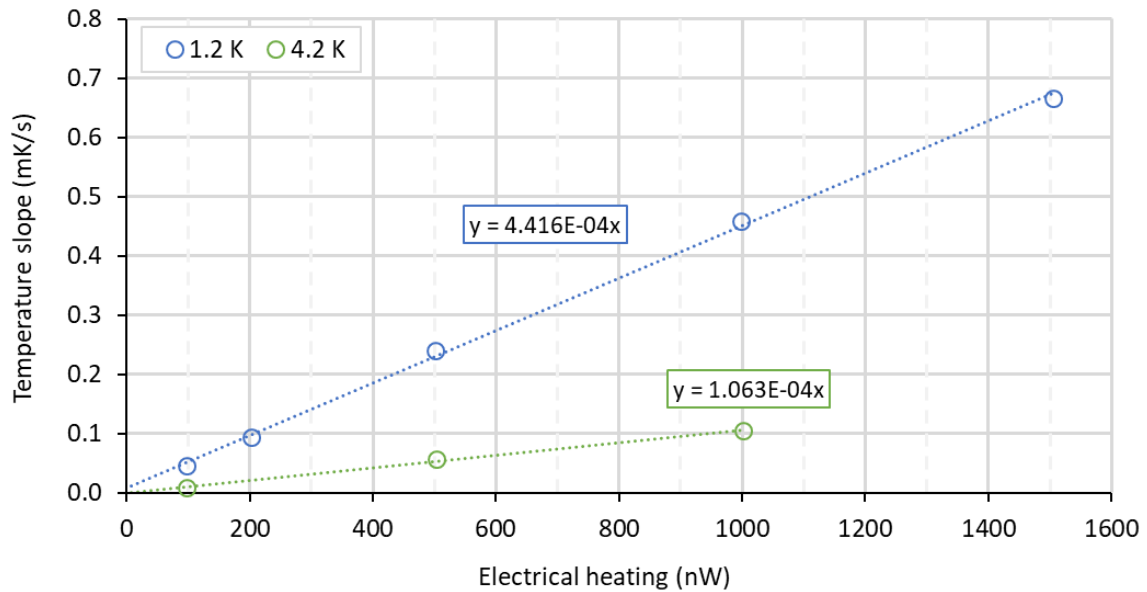


Figure 4.6 – Calibration curves for injected electrical heating at HS1 tip, at 1.2 K (blue) and 4.2 K (green)

The deduced sensitivities to dissipation (i.e. slopes of the calibration curves) are:

- at 1.2 K: 0.442×10^{-3} mK/s/nW
- at 4.2 K: 0.106×10^{-3} mK/s/nW

For given temperatures slopes such as those presented in Table 4.1 and Table 4.2, with these sensitivities we deduce the dissipation due to microvibrations at each excitation frequency, at a given temperature and shaker drive voltage level. The hypothesis is made here however, that dissipation occurs at the tip of the heat switch. As we shall see later, for a cantilever beam the validity of this assumption is debatable.

We also note that the sensitivity at 1.2 K is ~4.15 times higher than at 4.2 K. The next section explores the reason for this gain. We will then present the first dissipation curves.

4.1.4 Sensitivity gain 1.2 K versus 4.2 K

We can assume that the thermal behaviour of the heat switch is akin to a thermal RC circuit in series as described in the generalised thermal model of chapter 1. In this case the thermal capacitance is defined by the sum of the specific heats of each component at the heat switch tip, and the thermal resistance by that of the titanium tube. We showed that the initial temperature slope $\dot{T}(t \rightarrow 0)$ of the transient response, corresponding to the calculated temperature slopes presented here in section 4.1, is given by:

$$\dot{T}(t \rightarrow 0) = \frac{P}{C} \quad (4.1)$$

Where P would be the dissipation level arriving at the heat switch and C the total heat capacity of the heat switch tip. As we assume the dissipation level is independent of the working temperature, this suggests the factor ~4.15 gain in sensitivity observed is due to a corresponding drop in the capacitance.

In Figure 4.7 we show the evolution of the specific heat of the most common materials making up our test bench from 4.5 K down to 1 K: copper (Cu), titanium alloy (TA6V) and stainless steel (SS).

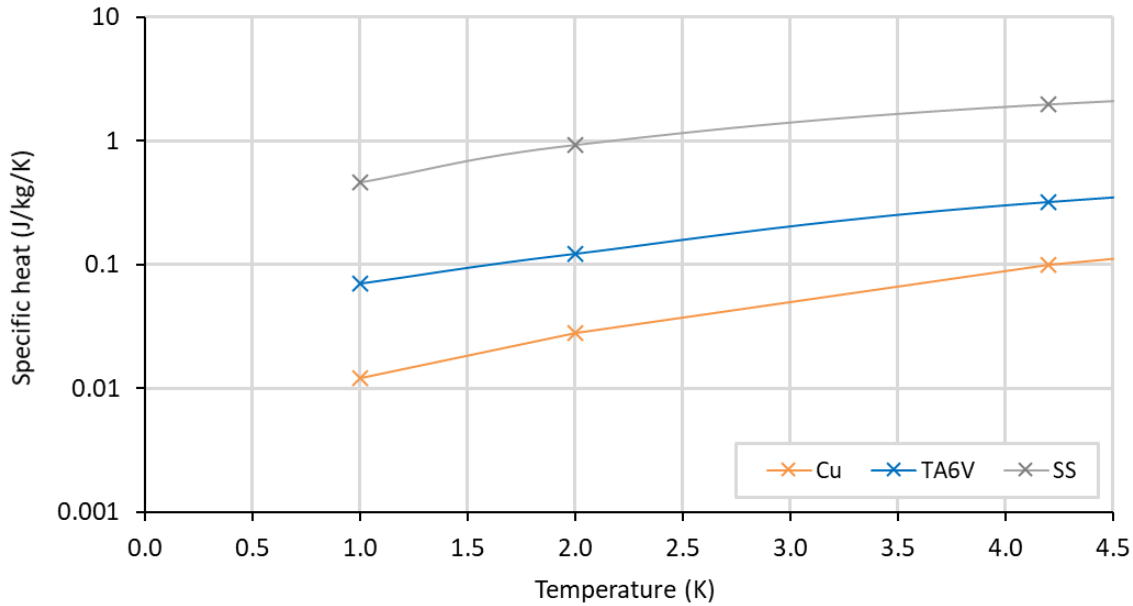


Figure 4.7 – Specific heat capacities of various materials below 4.5 K

We observe large changes of ~1 order of magnitude over just a few degrees, and large differences between materials, exceeding the order of magnitude between copper and stainless steel.

The total thermal capacitance at the tip of the heat switch is comprised of the copper tip itself, the instrumentation (thermometer and heater) and the stainless steel screws, nuts and washers that fix them together. Assuming the instrumentation has a capacitance similar to its mass in copper (it is difficult to calculate precisely), we can estimate the total capacitance of the heat switch tip.

Table 4.4 lists the heat capacities we use for our calculation, for copper and stainless steel at 4.2 K and 1.2 K. The rightmost column list the factor by which they fall.

Table 4.4 – Specific heat of copper and stainless steel at 4.2 K and 1.2 K

Material	$C_{p_4.2K}$	$C_{p_1.2K}$	$\frac{C_{p_4.2K}}{C_{p_1.2K}}$
<i>Copper</i>	$1.00 \times 10^{-1} \text{ J/kg/K}$	$1.47 \times 10^{-2} \text{ J/kg/K}$	6.8
<i>Stainless steel</i>	$1.96 \times 10^0 \text{ J/kg/K}$	$5.48 \times 10^{-1} \text{ J/kg/K}$	3.6

We recall from Table 2.1 the masses at the tip of the heat switch in Table 4.5.

Table 4.5 – Mechanical parameters of heat switch

	Parameter	Value
<i>Mass HS tip (copper)</i>	m_{tip}	11.9 g
<i>Mass instrumentation at tip (w/o steel)</i>	m_{instr}	4.5 g
<i>Screws, nuts and washers at tip (only steel)</i>	m_{steel}	2.6 g
<i>Mass total at tip</i>	$m = m_{tip} + m_{instr} + m_{steel}$	19.0 g

The total capacitance is therefore estimated with 16.4 g of copper and 2.6 g of stainless steel (totalling 19.0 g at the tip). The estimation of the total capacitance of the tip of the heat switch is then:

- at 4.2 K: $6.736 \times 10^{-3} \text{ J/K}$
- at 1.2 K: $1.666 \times 10^{-3} \text{ J/K}$

From 4.2 K to 1.2 K, calculated total capacitance decreased by a factor ~ 4.04 and the sensitivity to dissipation observed in section 4.1.3 increased by a factor ~ 4.15 . Our observations are therefore in close agreement with the prediction. The small difference could be explained by an imprecise assumption that the capacitance of the instrumentation is equal to its mass in copper.

As expected, from Table 4.4 we find that the sensitivity gain lies between the heat capacity ratio of copper (6.8) and stainless steel (3.6). It is closer to the ratio of steel because the heat capacity of steel is more than 1 order of magnitude greater than copper, as just discussed and shown in Figure 4.7. Hence despite the mass of steel being much lower than copper, its contribution to the total capacitance at the tip is larger.

4.1.5 Dissipation versus excitation frequency

When dividing the temperature slopes presented in Table 4.1 and Table 4.2 by the sensitivities given in section 4.1.3, we obtain the heat load due to microvibration dissipation arriving at the tip of the heat switch as a function of excitation frequency. This is shown in Figure 4.8, where a dissipation curve is plotted for each of the 3 shaker voltages (750 mV_{pp}, 500 mV_{pp} and 250 mV_{pp}) at 1.2 K and for 750 mV_{pp} at 4.2 K.

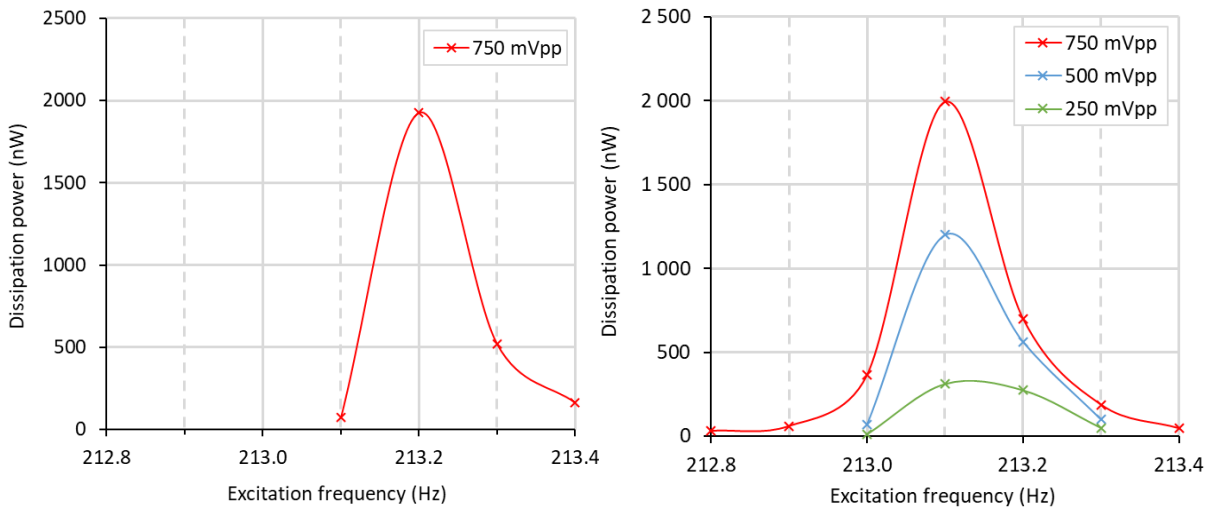


Figure 4.8 – At 4.2 K (left) and 1.2 K (right): microvibration dissipation at HS1 tip for several shaker drive voltages

As expected, at equal shaker drive voltage ($V_s = 750 \text{ mV}_{pp}$), the additional heat load at the tip of HS1 due to the dissipation of microvibrations at 4.2 K and 1.2 K is equal (here $2 \text{ } \mu\text{W}$). We also note that in the 1.2 K case (right side of Figure 4.8), lower shaker drive voltages lead to lower dissipation levels.

Regarding the mechanical resonance of heat switch HS1, it was found at 200 Hz using the laser sensor at room temperature (see section 2.2.4) and we note that the maximum dissipation is at 213.1 Hz. We are confident that this corresponds to the same natural frequency that has simply shifted upon cooling. Indeed, this is explained by increased mechanical stiffness (i.e. Young modulus) that usually occurs for metallic materials when cooled, as discussed in chapter 3 on the limitations of the measurement environment. As per Table 2.3, from 300 K to 4.2 K the Young modulus of the TA6V alloy used for the titanium tube of the heat switch increases by 12.7% (cooling further to 1.2 K only has negligible further impact on stiffness). This corresponds to a resonance frequency change of 6.2%, thus a shift from 200 Hz to 212.4 Hz. This is in close agreement with the maximum dissipation frequency of 213.1 Hz.

The slight shift of 0.1 Hz in the peak dissipation frequency from 4.2 K to 1.2 K seen Figure 4.8 cannot be explained by the same reasoning however. We note in this case that the frequency has *decreased* as temperature decreases, which is not in agreement with the evolution of the Young modulus with temperature. Furthermore, in this temperature range its change is in fact negligible. The shift could come from changes in the cryogen masses that as we have seen in chapter 3 affect mechanical behaviour.

The Q factor, a mechanical characteristic of the heat switch, can be deduced from the dissipation curves of Figure 4.8, using the bandwidth definition $Q = f_{res}/\Delta f$ discussed in chapter 1. The approximate values for the 4 dissipation curves are given in Table 4.6. We observe that the Q values are high (>1000).

Table 4.6 – Q factor values for dissipation curves of HS1 tip

	4.2 K	1.2 K
250 mV _{pp}	-	1000
500 mV _{pp}	-	1400
750 mV _{pp}	1600	1500

We should note that measured from the dissipation curves, these are *apparent* Q factors. Indeed, as discussed in chapter 3, the precision with which we measure the Q factor in this way is not always satisfactory. This is because the true acceleration profile is assumed flat, though in reality it is not. This distorts the resulting dissipation curve and so affects the Q factor calculation. For further reasons also outlined in chapter 3, the measurement of the dissipation curves themselves may also be imprecise.

In addition, the Q factors presented here were calculated (from the frequency-to-bandwidth ratio) using only the limited number of points available shown in Figure 4.8. They nevertheless indicate an order of magnitude of this mechanical characteristic.

It is interesting now, for each of the shaker drive voltages discussed, to look at the injected acceleration level (i.e. at the cold plate) across the frequency band where dissipations are observed.

4.1.6 Dissipation versus acceleration level

It is important to know the mechanical excitation (i.e. acceleration) level at the base of the heat switch that causes microvibration dissipation, rather than simply the shaker drive voltage. The injected acceleration level is measured using the cryogenic accelerometers placed as close as possible to the interface of the test object, here the base of the heat switch (see Figure 4.2). Indeed, as the shaker is mounted to the exterior of the cryostat, the true level of mechanical excitation seen by the test object is highly dependent on the cryostat mechanical transmission. This transmission is itself dependant on the excitation frequency and level as discussed in chapter 3, where the mechanical response of the cryostat was presented.

For the three shaker drive voltages of the microvibration tests presented (i.e. 250 mV_{pp}, 500 mV_{pp} and 750 mV_{pp}), acceleration measurements were carried out over the frequency band where microvibration dissipation was observed (212.8 Hz to 213.4 Hz). In these preliminary dissipation measurements, only the vertical (z) component of the injected acceleration was measured, as it was assumed the other components (x,y) were negligible. Acceleration and temperature measurements were also not synchronised at this stage. The dissipation curves at 1.2 K (right side of Figure 4.8) are shown again Figure 4.9, now with the corresponding vertical acceleration measurement at each excitation frequency.

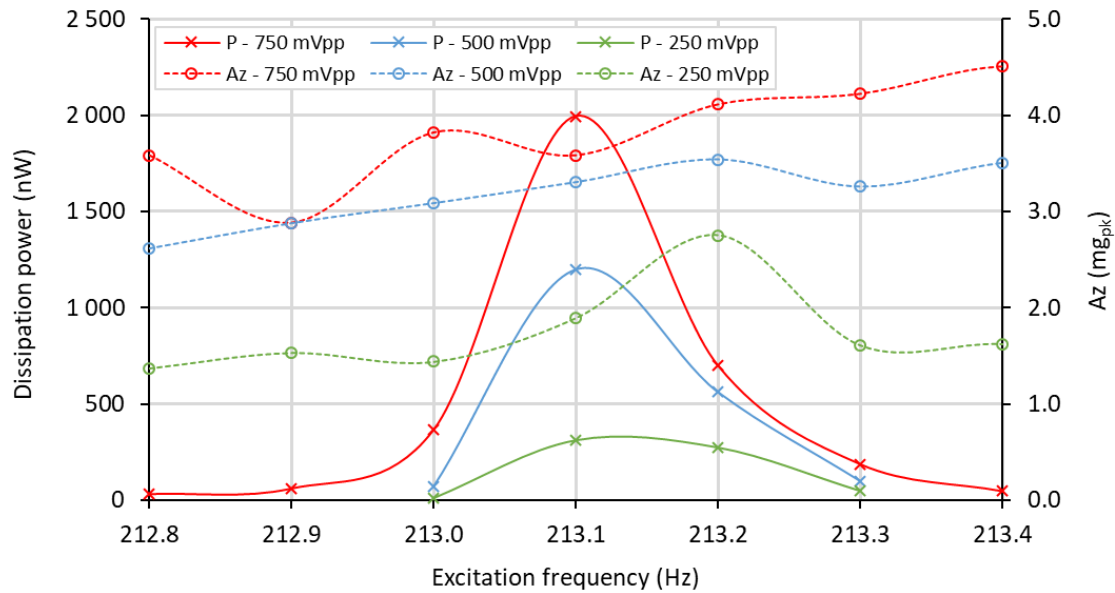


Figure 4.9 – Microvibration dissipation at HS1 tip and acceleration level, at fixed excitation frequencies, for several shaker drive voltages and at 1.2 K

We find that the acceleration levels are relatively stable in this frequency band, as opposed to the large variations of the cryostat's mechanical transmission observed for wide bandwidths as presented in chapter 3. We can therefore reasonably compare the average acceleration level across this narrow frequency band with the peak dissipation observed, for each shaker drive voltage. This is preferred over taking the acceleration level at the peak dissipation frequency (i.e. 213.1 Hz), as the acceleration and microvibration dissipation measurements are asynchronous.

Taking for each shaker drive voltage the average acceleration level and peak dissipation level, we obtain the points shown Figure 4.10. We note proportionally greater dissipation at increased acceleration levels. In fact, when fitting a power law passing through the origin, we obtain an exponent value of 2.73.

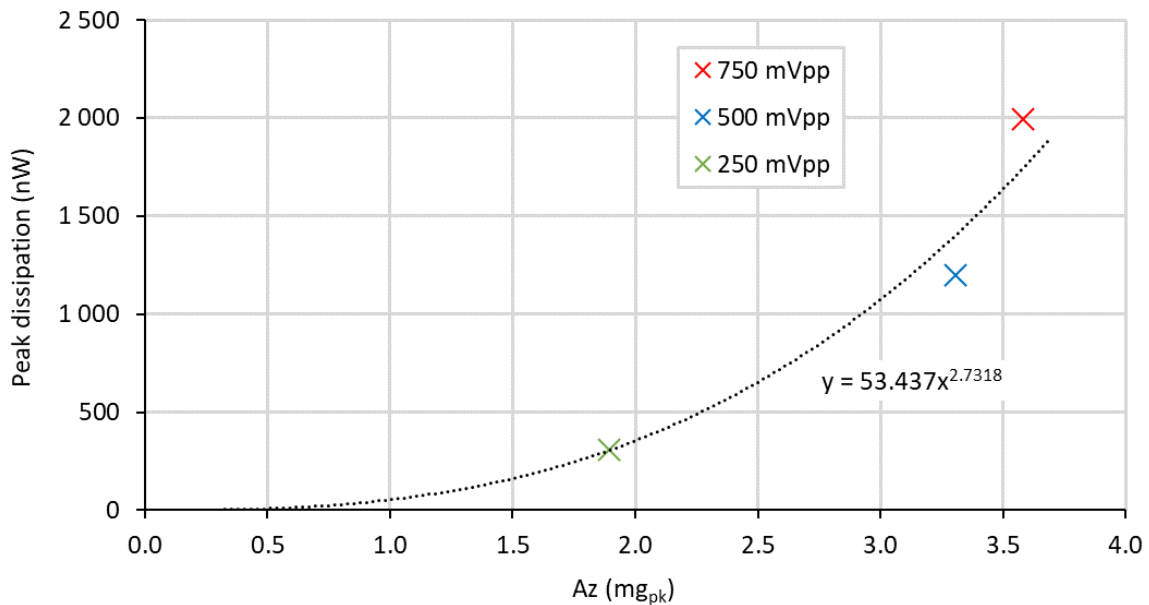


Figure 4.10 – Peak microvibration dissipation at HS1 tip as a function of average vertical acceleration level at 1.2 K

Referring back to the thermomechanical model of the damped mass spring, equation (1.45) predicts a proportionality between the peak dissipation P_{max} and the square of the injected acceleration level a (i.e. an exponent value of 2):

$$P_{max} = \frac{mQ}{2\omega_0} a^2$$

Though the preliminary dissipation measurements of Figure 4.10 only use 3 data points spread over a relatively narrow acceleration range to fit a power law, an exponent value of 2.73 instead of 2 suggested by equation (1.48), considering the experimental approximations, is compatible with this expression.

4.1.7 Correlation with model

It is also worth fitting the dissipation model presented in chapter 1 to the experimental data points. We take the experimental dissipation curve of the highest excitation level of Figure 4.9, and use as input to the model the measured parameters: $a = 3.6 \text{ mg}_{pk}$, $Q = 1500$ (as per Table 4.6), $m = 16.8 \text{ g}$ and $f_d = 213.1 \text{ Hz}$. Figure 4.11 shows the dissipation predicted by the model (dashed line) and the experimental data (solid line). The estimated peak dissipation is $11.7 \text{ } \mu\text{W}$, which can directly be calculated from equation (1.45). This overestimates the experimental data by a factor 5.9 ($11.7 \text{ } \mu\text{W}$ versus $2 \text{ } \mu\text{W}$).

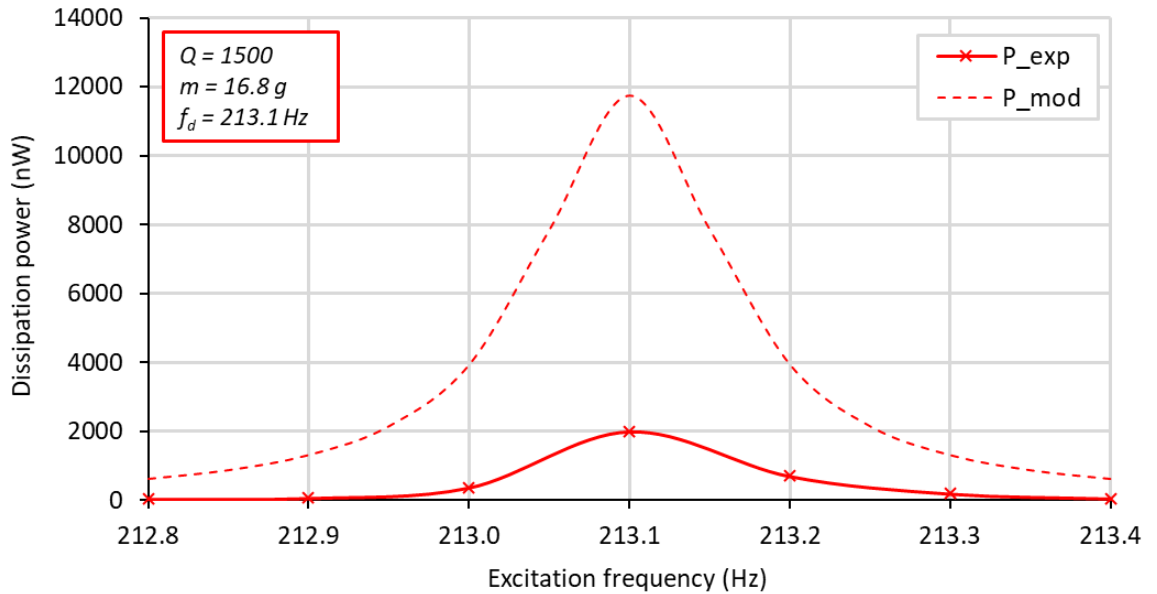


Figure 4.11 – Dissipation model (dashed line) and experimental data (solid line) for given fixed Q , m and f_d

The discrepancy between the model and the experimental data could be explained in several ways. Given the few experimental data points and imprecision with which the dissipation is measured, the estimation of Q (taken to be 1500 here) may also be imprecise. The true total mass at the tip of the heat switch may also be different, and there are uncertainties in the measured acceleration level. However, these aspects likely do not explain such a large factor 5.9 discrepancy.

It could however be explained by a recurring setup characteristic of this thesis: the measured dissipation is only a *part* of the total dissipation. If microvibration dissipation occurs elsewhere than at the heat switch tip, say in the titanium tube, at the base of the heat switch or elsewhere, only some of the resulting heat

load will arrive at the heat switch tip. The true dissipation level will therefore be underestimated, possibly by a significant factor. This notion will be explored further later in this chapter.

For convenience and to improve the readability of the plot (and an approach used later too), we fit the model to the measurements by introducing a scaling factor α that accounts for the overestimation of the model (or the underestimation of the measurements).

Adjustment of α can be done in 2 ways. The first is by fitting the peak dissipation levels and the second is by fitting the bandwidths (i.e. the full width at half-maximum, defined in section 1.3.7.1). We choose the former for all model correlations presented in this dissertation.

Thus here we set $\alpha = 5.9$. It is used as a reduction factor: if $\alpha > 1$ the model overestimates the dissipation level we believe is measured and if $\alpha < 1$ the model underestimates it. Figure 4.12 shows the estimated dissipation corrected using the scaling factor.

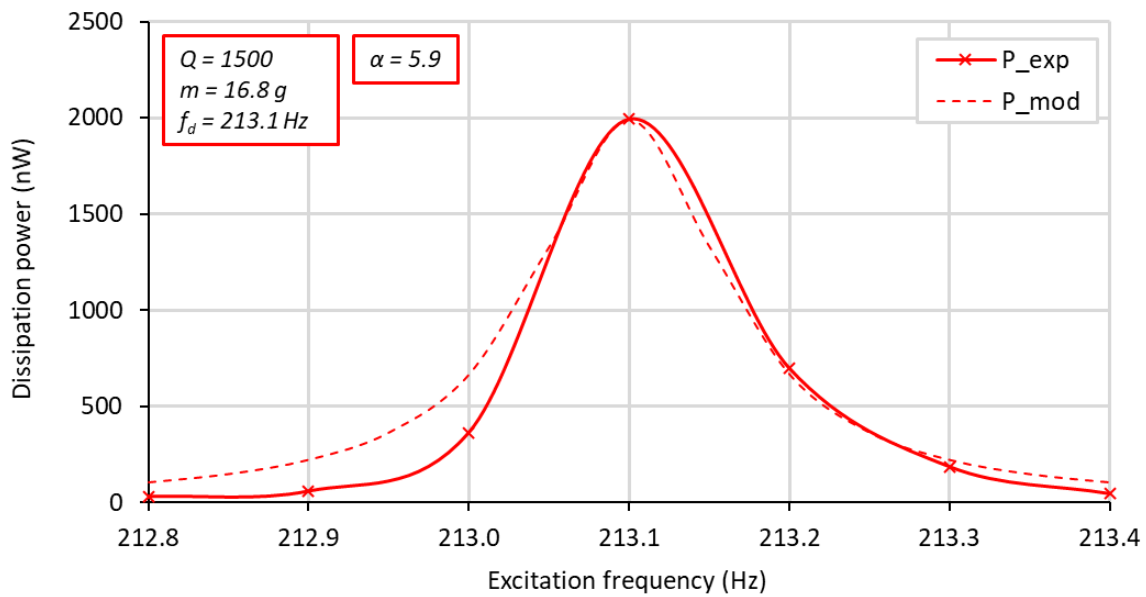


Figure 4.12 – Dissipation model (dashed line) fit to experimental data (solid line), using scaling factor α (fixed Q , m and f_d)

Assuming our model is physically representative of the dissipative behaviour of the heat switch and neglecting the approximations listed above, this scaling factor is an image of the proportion of microvibration dissipation that is unaccounted for in dissipation measurements.

4.1.8 Measurement difficulties

One important limitation of this setup is the rigid base configuration, implying that the dissipation occurring at the base of the heat switch cannot be measured. The analysis to overcome this difficulty is presented in detail in section 4.2. Prior to that, it is worth discussing the measurement difficulties encountered here during the preliminary measurements, which we attempt to mitigate with future test configurations. Despite these difficulties, the preliminary measurements presented here do still provide useful observations and reference values of resonances, acceleration levels and dissipation levels, to be compared to future tests.

4.1.8.1 Accelerometer dissipation

As discussed in chapter 2, the cryogenic accelerometers used are IEPE (Integrated Electronics Piezo-Electric) sensors that require a supply current, making them sources of joule heating. Throughout the experiments of this thesis, they were found to heat objects to which they were attached through conduction, as well as other nearby objects through radiation. Here we shall discuss the impact of this heating on experiments and how it was addressed.

The heat switches mounted as visible in the left side of Figure 4.2, we fixed 2 accelerometers between them onto the cold plate and orthogonally to each other (along y and along x). When mounted, they were slightly closer to HS1 (rear, with snubber) than HS2 (front, without snubber). The cryostat was cooled to 4.2 K such that the tips of the heat switches were insulated (i.e. in the OFF mode, where helium is trapped in the miniature sorption pumps, as described in section 2.2.1). The accelerometers were at first switched OFF (no supply current) and then switched ON at their nominal supply current of 3 mA each. This is the supply current recommended by the manufacturer to ensure a stable gain of the output sensor signal.

The evolution of the temperatures of the heat switch tips and the cold plate due to the joule heating caused by the accelerometers is shown Figure 4.13. Using the temperature sensitivity of HS1 deduced in section 4.1.3, we estimate the radiative power on the heat switches by measuring the temperature slopes.

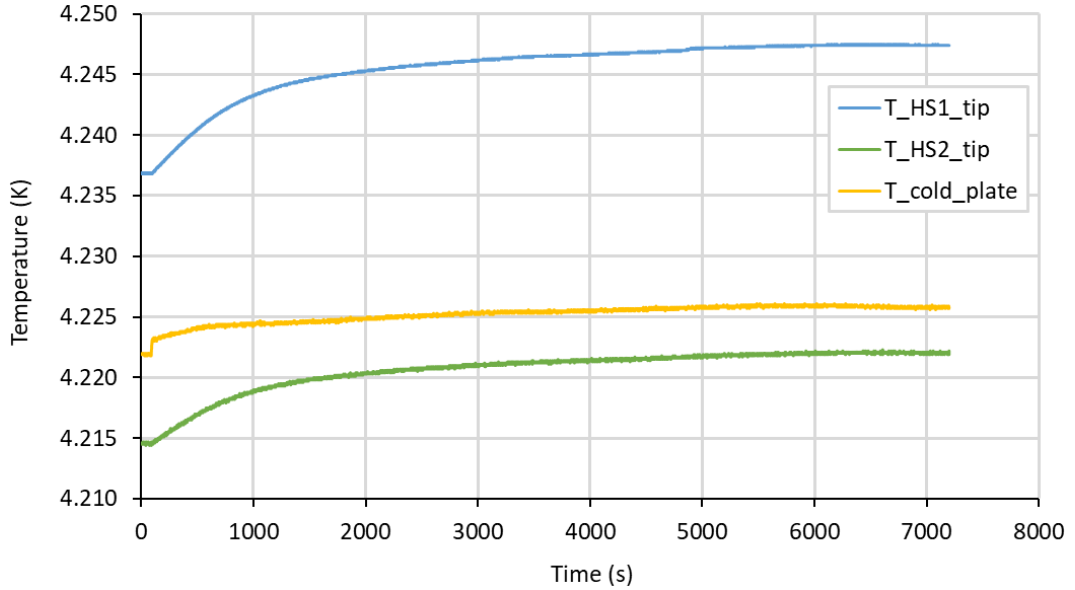


Figure 4.13 – Temperature elevations of heat switch tips and cold plate, due to switching ON of 2 accelerometers

We note that there is an offset between the thermometers, leading to a displayed tip temperature for HS2 that is lower than the cold plate. This is only an offset and does not hinder our analysis here.

The heat loads due to radiation from the accelerometers are, at the tips of HS1 and HS2, approximately 90 nW and 60 nW respectively (deduced by calibration using the electrical heaters at the tips of the heat switches). However, in the case of the temperature regulation method, this added radiative power does not affect the temperature of the tips. The regulation power simply reduces (an offset), which in fact poses no problem for our experiments, as long this radiative power remains constant.

Nevertheless, as a precaution, we explore the impact of reducing the supply current of the accelerometers to reduce their radiative power, as the levels involved are of the same order as the smallest microvibration dissipation levels we wish to observe. For this we chose an arbitrary constant excitation: a shaker drive voltage of $200 \text{ mV}_{\text{rms}}$ and an excitation frequency of 140 Hz. The resulting accelerometer output for a 3 mA supply current was taken as the 100% gain reference. The supply current was then gradually reduced through an adjusting screw on the signal conditioner, while maintaining said constant shaker excitation. The gain drop at 1.2 K with respect to the reference at 3 mA is shown in Figure 4.14.

Given the sharp drop below a supply current of 1.5 mA, we adjust it to this value. When a single accelerometer was turned ON at 1.5 mA, the radiative power at the tip of HS1 was now 15 nW. Due to a temperature measurement error with HS2, we could not deduce the radiative power at its tip in this case.

The key takeaway here is that a reduction of the supply current has reduced by a factor 6 the heating of the accelerometers. This means temperatures at the tip of a heat switch can be kept cooler for a given regulation power. Given the stabilisation times, we also take care not to carry out dissipation measurements too soon after the accelerometers are switched ON.

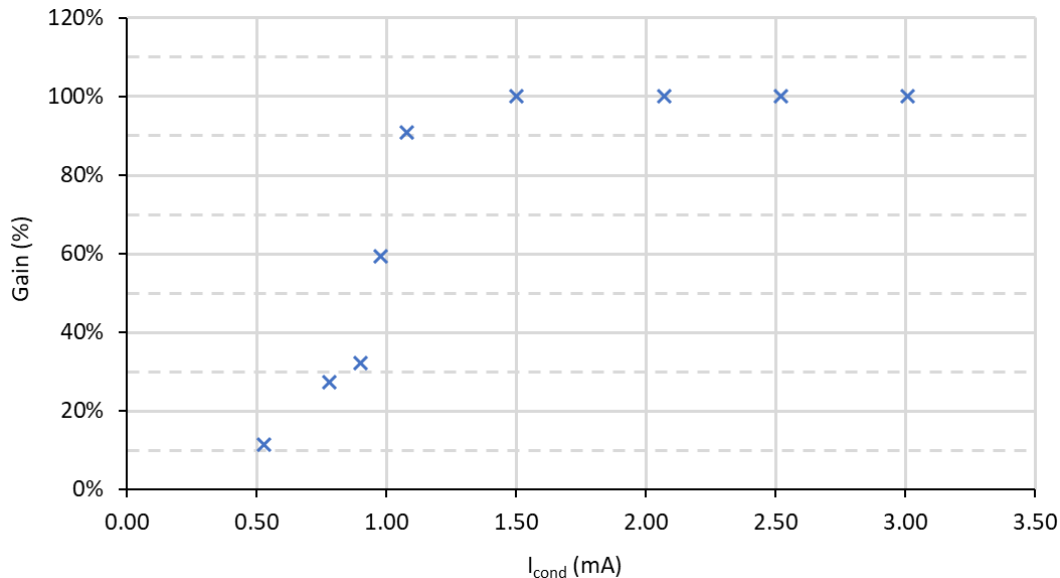


Figure 4.14 – Accelerometer + signal conditioner gain drop due to reduction of supply current at 1.2 K

In other tests we also found that the conductive heating of each accelerometer onto the cold plate was $\sim 10 \text{ mW}$ at this lower supply current (also deduced by calibration using the electrical heaters at the tips of the heat switches). When possible, we thus only switched those ON that were to be used, to avoid excessive heat loads on the heat switch tips but more importantly on the cold plate, to limit the boil-off rate of the helium bath (even if the impact is small given our cryostat losses of $\sim 200 \text{ mW}$).

4.1.8.2 Asynchronous measurements

Another key measurement difficulty during the preliminary measurements was that dissipation and acceleration measurements were not synchronous. As we have just seen, the accelerometers caused an additional heat load when switched on due to their supply current, both on the cold plate through conduction and at the tips of the heat switches through radiation. Upon this observation, we initially decided to carry out dissipation measurements independently of the mechanical ones. Another reason for

asynchronous measurements at the time of these measurements was a still decentralised LabVIEW program: one program performed temperature measurements and another acceleration measurements.

Asynchronous measurements would not be a concern if measurement repeatability was high. Particularly from a mechanical standpoint, the transmission of the cryostat is found to vary over time, as we have seen in chapter 3. We discussed how variations of the cryogen masses inside their respective reservoirs could alter the stability of the mechanical response of the cryostat. The result is that the exact acceleration level causing a given dissipation level is not confidently or accurately known, if these are not synchronised.

4.1.9 Summary

To conclude this section on the preliminary microvibration dissipation measurements of this thesis, we can summarise the key takeaways.

Given the measurement difficulties encountered during the preliminary dissipation measurements, the observations made must be taken with a pinch of salt. Thermal and mechanical measurements are asynchronous. The acceleration level is also underestimated as only the vertical component is measured. Lastly, the dissipation levels measured at the tip of the heat switch for these preliminary measurements are only a part of the total dissipation occurring in the heat switch. Keeping this in mind, the preliminary measurements presented in this section do nevertheless provide good insights.

First and foremost, heat loads due to the dissipation of microvibrations measured at the tip of HS1, were of the order of $2 \mu\text{W}$ for 4 mg of mechanical excitation in the range studied. This is a good reference figure for comparison to future tests. A finer interpretation is that we find proportionally greater dissipation at increased injected acceleration levels, not far from the square power law that the model predicts.

Next, a method of thermal measurement of the mechanical resonance of the heat switch has been validated. The measurement of microvibration dissipation as a function of excitation frequency has revealed itself a method of deducing said resonance, as well as of the Q factor, at cryogenic temperatures. The Q factors calculated from the dissipation curves were found to be very high (>1000), indicating dissipation over a very narrow frequency band (less than 0.6 Hz). In retrospect, this explains the difficulty with which we searched for the frequencies at which microvibration dissipations occurred.

To improve the quality of our experiments, we expect that decoupling the heat switch base from the cold plate will help. Detailed modelling of this is presented next.

4.2 Total dissipation and its location

Until now we assumed that the temperature elevation of the tip is a good estimation of the total dissipation in the heat switch. Here we explain why this is not necessarily the case and why we cannot currently identify the location of this dissipation either, justifying the need for new test configurations.

For the preliminary tests, the heat switch is mounted rigidly to the support bracket (see Figure 4.2). Together they can be considered a single element, both thermally and mechanically. From a thermal

standpoint this is limiting as any heat load arriving at the base of the heat switch (say from microvibration dissipation) cannot be measured because the cold plate will absorb it.

To support the analysis of our dissipation measurements, we propose a simplified thermal model of the heat switch. We will show that, to deduce the total dissipation occurring in the heat switch, the current configuration is limiting. New test configurations will better inform us on the total microvibration dissipation occurring in the heat switch, though its precise location still not.

4.2.1 Non-insulated heat switch base

Dissipation in the non-insulated heat switch base is presented here based on a simplified analytical model. The most basic representation of the heat switch when the base is not insulated, is shown Figure 4.15. This configuration is representative of the one used to carry out the preliminary dissipation measurements already presented. The heat switch base is thermalized to the cold plate and the low conductance titanium tube insulates the heat switch tip from it.

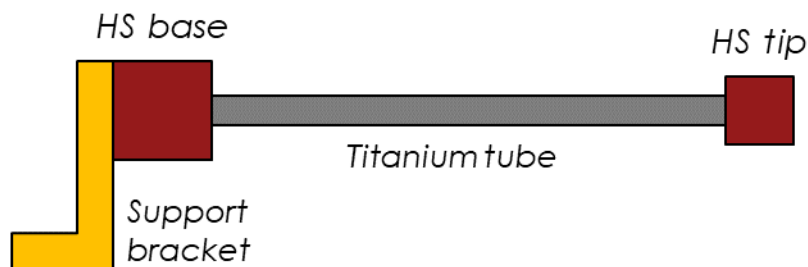


Figure 4.15 – Gas-gap heat switch thermal components

Before outlining the thermal model and its parameters, we make several hypotheses regarding the thermal, as well as mechanical, behaviour of the heat switch.

4.2.1.1 Hypotheses

Firstly, we recall the generalised thermal model of section 1.4, where we describe a simple thermal RC circuit in series. The heat switch in this configuration, its thermal behaviour can be approximated by considering the titanium tube as a pure thermal resistance R and the heat switch tip as a pure heat capacitance C . As we are only interested in the thermal equilibrium – and having concluded in section 1.4 that the thermal equilibrium of an RC circuit is independent of C – we can simplify the description of the thermal behaviour of the heat switch by simply modelling it as a thermal resistance.

Modelling the thermal behaviour of the heat switch in this way implies that conduction is the primary heat transfer mechanism. In the vacuum environments in which we work, convection is absent. At low temperatures (< 5 K), for this heat switch design geometry and small temperature differences between neighbouring objects, thermal radiation is also negligible compared to conduction effects.

Another hypothesis concerns the mechanical behaviour of the heat switch, in attempting to predict dissipation profiles along its length. It is assumed its behaviour is akin to a mechanical cantilever beam.

Lastly, the heat dissipation it experiences and the imposed temperature of its base (the boundary condition) will define the thermal equilibrium of the heat switch. We assume that the boundary condition

is the temperature of the cold plate, being thermalized to the base. In practice, this temperature is imposed at 4.2 K or 1.2 K by the helium bath.

4.2.1.2 Model outline

We attribute a first set of thermal parameters to this non-insulated base configuration. These are the conductance between the base and tip G_{12} , and the temperatures of the base T_1 and tip T_2 . They are shown in Figure 4.16. Subscript “1” refers to the heat switch base and subscript “2” to the heat switch tip.



Figure 4.16 – Gas-gap heat switch thermal parameters

The boundary condition being the temperature of the cold plate, it is represented here by T_1 .

Assuming negligible thermal contact resistance of the interface between the titanium tube and the copper components (silver brazing), the conductance G_{12} [W/k] of the titanium alloy tube is a function of the conductivity of titanium $k(T)$ [W/m/k], its cross-sectional area S [m²] and length L [m]:

$$G_{12} = \frac{S}{L} \cdot \frac{1}{T_1 - T_0} \int_{T_0}^{T_1} k(T) dT \quad (4.2)$$

In this form, we can only inject power at the base and tip of the heat switch. To inject power at intermediate locations, we must discretise the heat switch along its length. We can then deduce the thermal equilibrium.

4.2.1.3 Model solution

The heat switch of length L is discretised into N nodes joined by $N - 1$ elements. Each node n has a temperature T_n and each element a conductivity k_n . These parameters are depicted Figure 4.17.

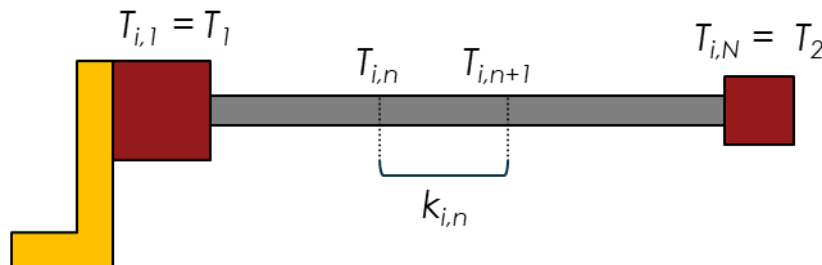


Figure 4.17 – Gas-gap heat switch iterative model parameters

A heat dissipation profile is constructed by injecting different power levels at individual nodes along the length of the heat switch. Heat dissipation injected at a node n is denoted P_n . The thermal equilibrium of the element between node n and node $n + 1$ for all $N \geq n \geq 1$, at iteration i , is given by:

$$(T_{i,n+1} - T_{i,n}) \frac{k_{i,n} S}{L/(N-1)} = \sum_{j=n}^N P_{i,j+1} \quad (4.3)$$

The temperature of each node depends on the temperatures of the nodes on either side of it and the thermal conductivities between them, themselves a function of the node temperatures. The conductivity $k_{i,n}$ of the element between node n and node $n + 1$ at iteration i , is deduced from the two respective node temperatures at the previous iteration $i - 1$:

$$k_{i,n} = k \left(\frac{T_{i-1,n+1} + T_{i-1,n}}{2} \right) \quad (4.4)$$

It is interesting now to explore the limitations of the current heat switch configuration through analysis of the model solution. We will first show how knowledge only of T_2 does not allow us to deduce the total dissipation in the heat switch, by studying power injected at individual nodes and at different locations.

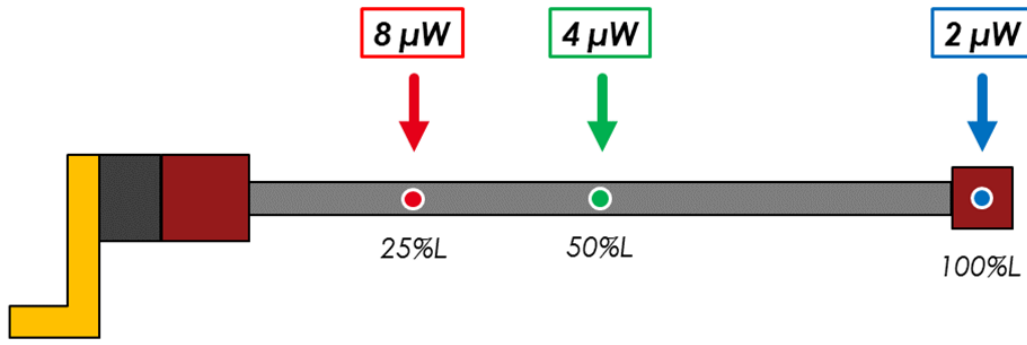
4.2.1.4 Point power dissipation

The simplest way to load the thermal model is to inject all the dissipation at a single node. We will refer to this as point power dissipation. The purpose here is to show the impact of different dissipation levels at different locations on the resulting temperature profiles and observed T_2 values.

As in the first configuration the heat switch base (T_1) is thermalized to the cold plate, only T_2 changes when dissipation occurs. We can study the validity of the calibration method described in section 4.1.3 by comparing cases of different point power dissipations injected at different locations (i.e. nodes).

We impose increasingly higher power levels for a node that is further away from the heat switch tip, aiming for a similar temperature elevation response at the tip. Indeed, we expect the impact of a given dissipation level on T_2 to be reduced the further it is from the tip. We study 3 cases (depicted in Figure 4.18):

- Case #1: $2 \mu W$ at 100% L (= 50 mm) *at heat switch tip*
- Case #2: $4 \mu W$ at 50% L (≈ 25 mm) *at midway point*
- Case #3: $8 \mu W$ at 25% L (≈ 12.5 mm) *near heat switch base*

Figure 4.18 – Power dissipation load cases (2 μW , 4 μW and 8 μW)

Imposing the boundary condition $T_1 = 4.2 \text{ K}$, the temperature profile for each case is shown Figure 4.19.

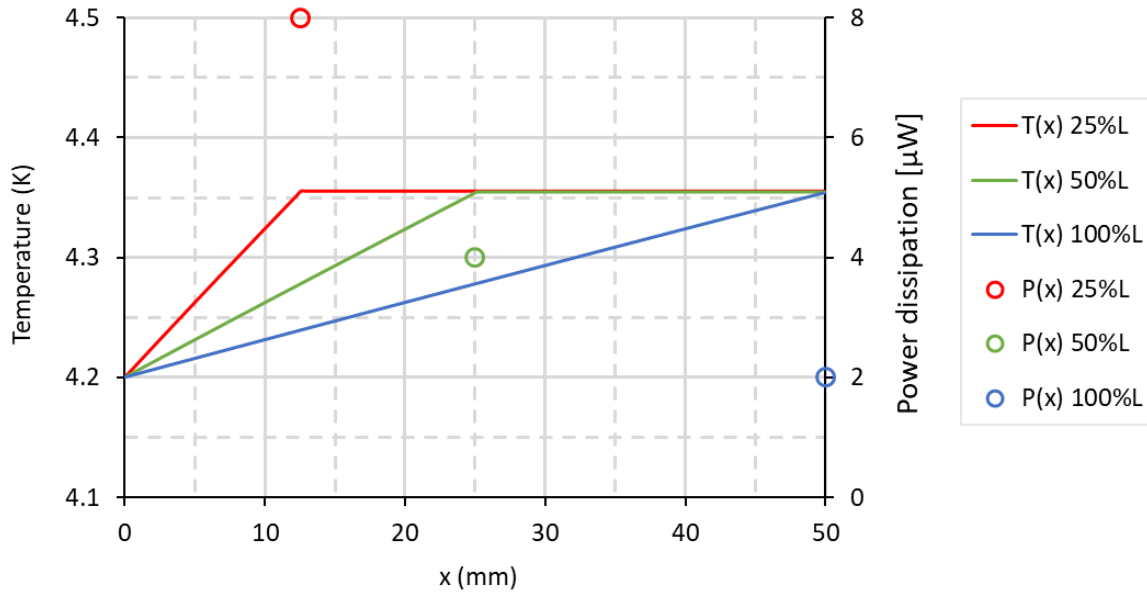


Figure 4.19 – Temperature profiles for different point dissipations at different locations

We note that the heat switch tip temperature T_2 (at $x = 50 \text{ mm}$) is the same in each case, namely a temperature elevation $\Delta T = T_2 - T_1 = 155 \text{ mK}$. We observe an inverse proportionality between the distance of the dissipation from the base of the heat switch and the magnitude of the point dissipation needed at that distance to maintain a constant T_2 value. This is because given the small temperature gradients here, k is almost constant with temperature. This means equation (4.3) be solved analytically without needing to discretise and solve by iteration with equation (4.4).

We therefore conclude here that a given elevation of T_2 is not a unique image of the dissipation occurring in the heat switch. Knowledge solely of the temperature elevation at the tip does not allow us to deduce the total dissipation occurring in the heat switch nor its location. To address this, section 4.2.2 will now cover how the test configuration is adapted by insulating the heat switch base

4.2.2 Insulated heat switch base

If the base of the heat switch is insulated from the cold plate, measuring its temperature with respect to the cold plate will allow us to deduce the power flowing from the base to the cold plate. In addition to the thermal parameters presented in section 4.2.1.2, we introduce the conductance between the bracket and

the base G_{12} and the temperature of the bracket T_0 that is still thermalized to the cold plate. The full set of thermal parameters for the insulated base is shown in Figure 4.20.

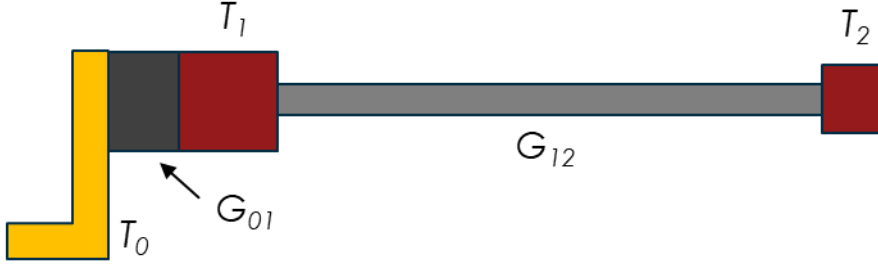


Figure 4.20 – Gas-gap heat switch with insulated base thermal parameters

4.2.2.1 Deducing total dissipation

As we have assumed that all dissipation in the heat switch is evacuated through its base (no convection, radiation or other conduction paths), through the measurement of T_1 versus T_0 we have at our disposal a heat switch configuration that allows us to deduce the total microvibration dissipation in the heat switch:

$$P_{total} = (T_1 - T_0) \cdot G_{01} \quad (4.5)$$

4.2.2.2 Deducing dissipation location

An evolution to the goal of deducing the total dissipation in the heat switch is to know its location. We construct different dissipation profiles along the length of the heat switch, by extending the point power dissipations presented in section 4.2.1.4 to each node. In the case of dissipation profiles, we will refer to the location of dissipation as its centroid position, that is to say the location where a single point power dissipation having the same total dissipation has the same impact on the temperature T_2 .

As we have now insulated the heat switch base, we know the total dissipation in the heat switch P_{total} , as well as T_2 . Assuming constant conductivity k , we can then express T_2 in terms of P_{total} and the dissipation position x_c from the base of the heat switch:

$$(T_2 - T_1) \frac{kS}{x_c} = P_{total} \quad (4.6)$$

Making several further assumptions regarding dissipation along the heat switch, we will now explore whether knowledge of the dissipation location can inform us on the shape of the dissipation profile.

4.2.2.3 Deducing dissipation profile

We will now study the shape of the dissipation profile using a mathematical approach. We recall that it is assumed the mechanical behaviour of the heat switch is akin to a cantilever beam. The bending stress σ along the length x of a cantilever beam with a point load F at its end, is given by:

$$\sigma(x) = \frac{-F \cdot (L - x) \cdot y}{I}$$

Parameter y is the perpendicular distance from the neutral fibre and I the moment of inertia of the beam cross section. Further assuming solid damping (see section 0), we expect dissipation to be greatest where stress is greatest (at the base of the heat switch) and zero at its tip (where stress is absent). Accounting for these assumptions, if dissipation is proportional to bending stress the dissipation *density* profile (in W/m) takes on the form, for A constant:

$$P(x) = A(L - x) \quad (4.7)$$

To widen our scope, we also decide to study power law dissipation density profiles of order $n \geq 0$:

$$P(x) = A(L - x)^n \quad (4.8)$$

The total dissipation in the heat switch P_{total} (in W) is obtained by integrating the dissipation density profile along the length of the heat switch:

$$P_{total} = \int_0^L P(x) \cdot dx = A \frac{L^{n+1}}{n+1} \quad (4.9)$$

We can also express the dissipation location x_c of a dissipation profile $P(x)$ in terms of L and n , from the definition of the centroid position of a function:

$$x_c = \frac{\int_0^L x \cdot P(x) \cdot dx}{\int_0^L P(x) \cdot dx} = \frac{\int_0^L x \cdot P(x) \cdot dx}{P_{total}} = \frac{L}{n+2} \quad (4.10)$$

Thus, given the assumptions made regarding the dissipation density profile, a pair of T_2 and P_{total} values and a boundary condition T_1 necessarily define a dissipation location x_c , as per equation (4.6). We then also deduce the order n of the dissipation density profile as per equation (4.10).

Imposing P_{total} we can show the relationship between n and the tip temperature T_2 . In Figure 4.21 we show the dissipation profiles for $n = 0$ (flat), $n = 1$ (linear), $n = 2$ (quadratic) and $n = 3$ (cubic), and the resulting temperature profiles, along the heat switch. We impose a total dissipation of 8 μ W to each profile. As n is defined, the centroid position x_c is too. As these differ, so too do the T_2 values (at $x = 50$ mm).

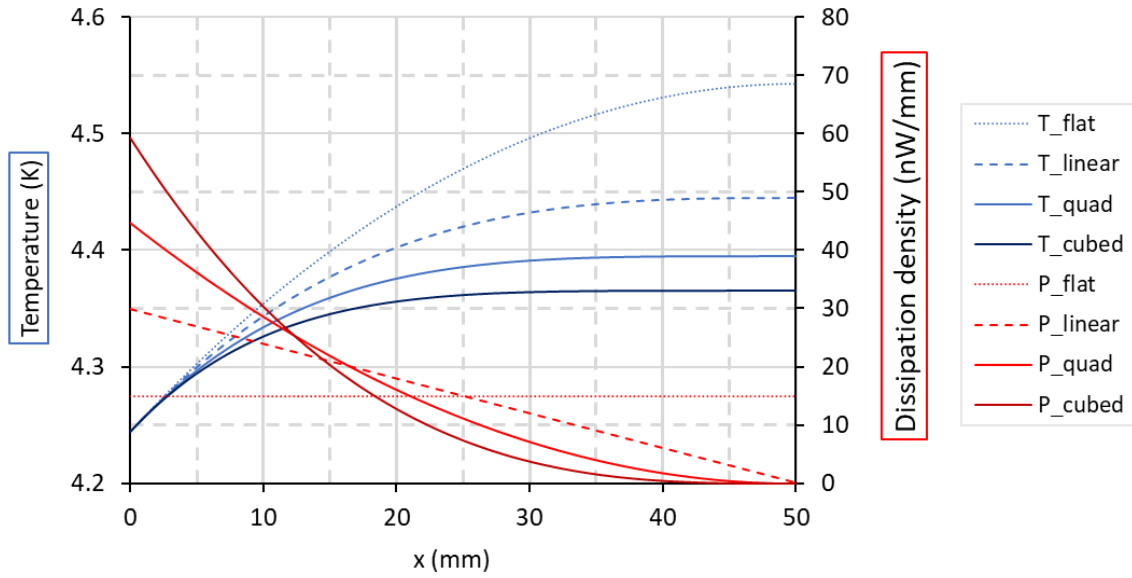


Figure 4.21 – Different dissipation density profiles and resulting temperature profiles (same total dissipation of 8 μ W)

It must be noted though, that the above relationships between P_{total} , T_2 (as inputs) and x_c , n (as outputs), have only been shown to be true here for dissipation density profiles described by a power law and having assumed zero dissipation at the tip of the heat switch. As discussed in Appendix 3, even a small heat load arriving at the tip drastically increases the deduced order of the dissipation profile. As a consequence, we are still limited by our setup in that we are unable to fully evaluate the dissipation profile.

4.2.3 Impact on experiments

It is now worth summarising the takeaways of section 4.2 and discussing the impact on future experiments. We have discussed how to deduce the total dissipation in the heat switch, its “average” location and the shape of its profile. A simple thermal R circuit was used to show that with the current heat switch configuration we cannot correctly evaluate dissipation due to microvibrations, justifying the need for more advanced test configurations. These are presented in section 4.3.

It was demonstrated that a given elevation of T_2 was not a unique image of the total dissipation. In other words, an infinite number of dissipation profiles along the titanium tube can lead to the same temperature elevation at the tip. We propose to insulate the base of the heat switch, allowing the measurement of T_1 to deduce total dissipation. The lower the conductance between the bracket and base G_{01} , the greater the temperature difference $T_1 - T_0$ for a given total dissipation, and so the more finely it can be resolved.

Insulating the heat switch base was shown, under certain hypotheses, to inform us on the dissipation location. By measuring the total dissipation, and the temperatures of the tip and base, we can deduce a unique dissipation *centroid position* (i.e. its “centre of mass”). If dissipation is located at a single point this directly corresponds to the dissipation location. If dissipation follows a power law profile along the heat switch, from this centroid position we can deduce the order of said power law.

The caveat however, is that this is only true if the dissipation profile is indeed described by a power law and that no dissipation occurs at the tip of the heat switch, such that it has the form given in equation

(4.8). Indeed, despite insulating the heat switch base, we are unable to differentiate between types of dissipation profiles (say for example power law versus exponential).

As we shall see, tests carried out with the advanced test configurations did in fact show dissipation arriving at the tip of the heat switch. Despite insulating the base, we therefore cannot confidently evaluate the dissipation profile. Ideally, a continuous measurement of the temperature along the length of the titanium tube would inform us on the temperature profile and so the dissipation profile.

In conclusion, insulating the heat switch base will allow us to deduce the total microvibration dissipation occurring in the heat switch as well as its location, though will not allow us to confidently evaluate the dissipation profile of the heat switch (i.e. along the titanium tube) as it is currently instrumented.

4.3 Advanced test configurations

The insulation of the heat switch base will now allow the measurement of the total microvibration dissipation in the heat switch, which was previously not possible. It will also help identify the location (or locations) of microvibration dissipation.

Now we will present the different insulated base configurations studied, followed by their mechanical and thermal characterisation. We will then discuss other setup and test protocol improvements. Lastly, we will present our first observations with the advanced test configurations and some side effects observed.

4.3.1 Insulated base configurations

4.3.1.1 Target conductance

The base of the heat switch is now to be thermally decoupled from the support bracket, yet still mechanically coupled such that microvibrations can be induced. For sizing the insulating components we can aim for a conductance G_{01} relative to the conductance G_{12} of the titanium tube. These parameters were previously described in section 4.2.2 and are labelled in Figure 4.20.

We recall that lower conductivity is preferable as it improves the thermal susceptibility of the test object to a heat load. The trade-off however is that a G_{01} value that is too low (i.e. high thermal resistance) increases the cooling time of the new test configuration. The heat switch itself circumvents this trade-off as it can be switched ON (high conductance) for cooling and then switched OFF (low conductance) for high thermal susceptibility during microvibration dissipation measurements.

A G_{01} value that is too high however, will not insulate the base of the heat switch sufficiently: high regulation power would be needed to create only a small temperature difference with respect to the cold plate, as per equation (4.5). This should also be avoided as it causes noisier regulation power, as well as big temperatures differences in the test object and so unaccounted for changes in material properties.

For these reasons, we choose to aim for a conductance between the bracket and base that is approximately 1 order of magnitude higher than the conductance of the titanium tube:

$$G_{01} = 10 * G_{12} \quad (4.11)$$

As discussed in section 2.2.1 when presenting the heat switch, at 4.2 K the conductance between the tip and the base when in the OFF mode (i.e. of the titanium tube) is rated at 12 $\mu\text{W/K}$. As we prefer to work at 1.2 K for improved bath temperature stability (see section 3.1.5) and higher sensitivity still, we must estimate this OFF-mode rated conductance at this lower temperature.

The conductivity of the titanium grade used (TA6V) decreases by a factor 5 from 4.2 K to 1.2 K. This is shown in Figure 4.22. The estimated G_{12} conductance in the OFF mode at 1.2 K is thus $\sim 2.4 \mu\text{W/K}$.

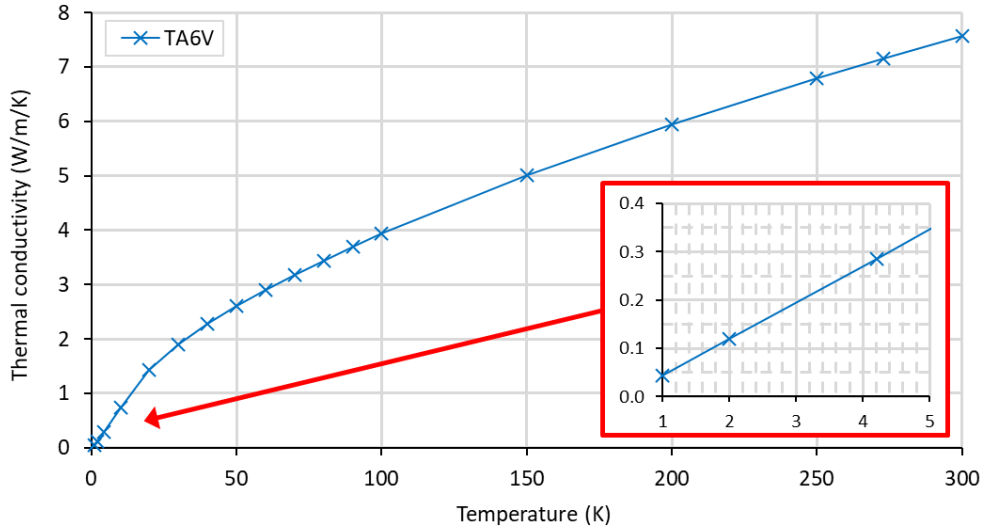


Figure 4.22 – Thermal conductivity of titanium alloy (TA6V) between 0 K and 300 K

The target G_{01} value is therefore $\sim 24 \mu\text{W/K}$ at 1.2 K. This is the total conductance between the bracket and heat switch base. To estimate it, we must consider the various conductances between the bracket, base and screw assemblies. The adopted configuration is designed to reduce these conductances.

4.3.1.2 Adopted configuration

The proposed configuration to thermally insulate the heat switch is shown schematically in Figure 4.23. The insulating elements used are spacers (between the bracket and base) and washers (between the base and screw heads). In total, there are 8 insulating elements (2 per screw assembly).

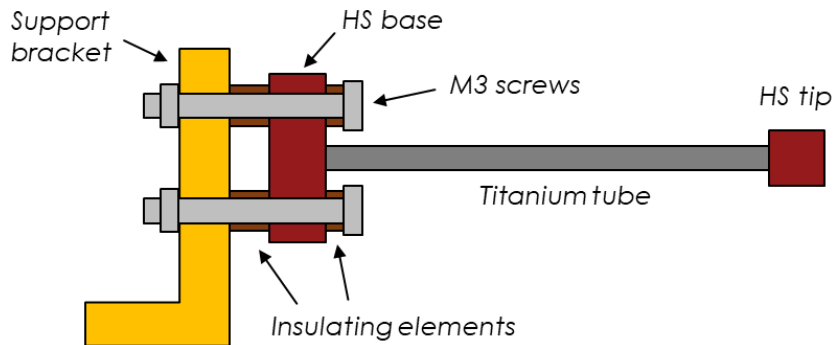


Figure 4.23 – Heat switch mounted with M3 screws (insulated base configuration)

For sizing the insulating elements, the 2 thermal paths must be accounted for. Regarding the first thermal path – the thermal contact resistance (TCR) between the bracket and base – the contact surface between the two components being the greatest and both made of copper, it is assumed the conductance between them is high compared to other conductances. Hence we first add insulation directly between the bracket and base. Regarding the second thermal path – through the screws – whose conductance is now non-negligible, insulation must be added between the bracket and screws or between the screws and base.

The first option thermalizes the screws to the base, whereas the second option thermalizes the screws to the bracket. The latter is preferred as thermal sensitivity to microvibration dissipation was shown to be inversely proportional to heat capacitance (when measured dynamically): to maximise sensitivity, the thermal capacitance of the base should be minimised.

Analytically the total conductance G_{01} (ignoring the TCRs, which is difficult) is the sum of the conductances of the parallel thermal paths:

$$G_{01} = 4 * G_{spacer} + 4 * \left(\frac{1}{G_{washer}} + \frac{1}{G_{screw}} \right)^{-1} \quad (4.12)$$

For a single screw assembly, the insulating washer and screw are thermally in series, thus the thermal resistances $R = 1/G$ must be summed. This corresponds to the term in brackets of equation (4.12). Next, the 2 thermal paths (through the spacers or through the screws and washers) are thermally in parallel, thus can be summed directly. Lastly, the factor 4 reflects that we have 4 screw assemblies in parallel.

Next, given equation (4.2), we must impose dimensions (cross-sectional area and length) of the screws and insulating elements to achieve a desired G_{01} value. The effective length of the thermal path through the screws is imposed by the lengths of the insulating elements. It is equal to the sum of the thicknesses of the bracket, base, insulating spacer and insulating washer. This is visible Figure 4.23.

In the lab where the insulating elements were machined, several material choices were available. They were compared from both thermal and mechanical perspectives and will now be presented.

4.3.1.3 Composite material insulation

A selection of low conductivity materials was made from those available to us in the lab. We first consider 2 materials often used in cryogenics for insulation that show good mechanical strength: G10 (a fiberglass epoxy laminate) and CFRP (carbon-fibre reinforced polymer). For our application we also consider 2 additional materials, specifically 2 grades of the trademarked polyimide plastic Vespel: SP1 (pure polyimide resin) and SP22 (polyimide with 40% graphite). Both display high thermal insulation and good mechanical strength, though SP1 is more insulating and mechanically stronger. According to in-house data, the conductivities of these 4 materials, both at 1.2 K and 4.2 K are summarised Table 4.7.

These figures must be considered with caution. Both G10 and CFRP are anisotropic by nature, due to fibre direction. This translates to thermal conductivities that vary depending on the direction of heat flux relative to these fibres. Table 4.7 only aims to give an idea of orders of magnitude.

Table 4.7 – Conductivities of material selection for heat switch insulating elements

	Conductivity at 1.2 K	Conductivity at 4.2 K
<i>G10 (in-plane)</i>	2.10×10^{-2} W/m/K	6.33×10^{-2} W/m/K
<i>CFRP (in-plane)</i>	9.26×10^{-3} W/m/K	2.88×10^{-2} W/m/K
<i>Vespel SP22</i>	2.38×10^{-3} W/m/K	2.42×10^{-2} W/m/K
<i>Vespel SP1</i>	2.24×10^{-3} W/m/K	1.01×10^{-2} W/m/K

We observe that the 2 grades of Vespel are considerably more insulating than G10 and CFRP. At 1.2 K (the temperature at which we will most often work) we note only a negligible difference between the SP22 and SP1 grades. Given the rods and tubes available in the lab that will be used to machine the insulating elements, it was decided to machine sets in 2 materials: Vespel SP22 and CFRP. The former shows very low conductivity at 1.2 K, while the latter is commonly used for its high mechanical stiffness. We will later compare the sets both thermally and mechanically.

Regarding dimensions of the insulating elements, we recall that they are chosen aiming for a G_{01} value of ~ 24 μ W/K at 1.2 K. Several constraints exist when defining these dimensions. The inner diameter of the insulating elements must be greater than 3 mm as they are mounted around M3 screws. Their thickness must also be sufficiently high for mechanical strength, though not too high for good thermal insulation. The opposite applies to the chosen length, which must be sufficiently short for mechanical strength, though not too short for good thermal insulation.

The dimensions of the CFRP tubes at our disposal also limit our choices. As it is a composite material that is not easily machined, the inner and outer diameters are imposed by the CFRP tube available. Vespel however has no such constraints, giving us more freedom in the dimensions. Given the target conductance, the material conductivities (Table 4.7) and the CFRP tube constraints, the final dimensions of the insulating elements are given in Table 4.8.

Table 4.8 – Dimensions of insulating elements in CFRP and Vespel SP22

	CFRP	Vespel SP22
<i>Inner diameter</i>	4.0 mm	3.2 mm
<i>Outer diameter</i>	5.0 mm	5.2 mm
<i>Spacer length</i>	10 mm	10 mm
<i>Washer length</i>	2 mm	2 mm

The machined insulating elements are shown for each composite material in Figure 4.24.



Figure 4.24 – Composite material insulating spacers and washers, in CFRP (left) and Vespel SP22 (right)

Using equation (4.2), from Table 4.7 and Table 4.8 we can estimate the conductances of the insulating elements for each material. These are summarised for 1.2 K in Table 4.13.

Table 4.9 – Theoretical conductances at 1.2 K of insulating elements in CFRP and Vespel SP22

	CFRP	Vespel SP22
$G_{spacer} \text{ at } 1.2 \text{ K}$	6.5 $\mu\text{W/K}$	3.1 $\mu\text{W/K}$
$G_{washer} \text{ at } 1.2 \text{ K}$	32.7 $\mu\text{W/K}$	15.7 $\mu\text{W/K}$

Given the dimensions of Table 4.8 and knowing that the thickness of the bracket is 8 mm and the thickness of the base is 8.5 mm, we also deduce an effective screw length of 28.5 mm (8+8.5+10+2). Together with the cross-sectional area of an M3 screw and the conductivity of stainless steel, the conductance of one M3 screw is estimated. These figures are summarised Table 4.10.

Table 4.10 – Theoretical conductance at 1.2 K and 4.2 K of stainless steel M3 screw

M3 steel screw	
$k_{steel} \text{ at } 1.2 \text{ K}$	$5.27 \times 10^{-2} \text{ W/m/K}$
$k_{steel} \text{ at } 4.2 \text{ K}$	$2.54 \times 10^{-1} \text{ W/m/K}$
S_{screw}	7.1 mm ²
L_{screw}	28.5 mm
$G_{screw} \text{ at } 1.2 \text{ K}$	13.1 $\mu\text{W/K}$
$G_{screw} \text{ at } 4.2 \text{ K}$	63.0 $\mu\text{W/K}$

Finally, from equation (4.12) we deduce the total conductance G_{01} of the 8 insulating elements (4 spacers and 4 washers) for both the CFRP set and the Vespel set, at 1.2 K. The same calculations are also done at 4.2 K for comparison. The deduced G_{01} values at 1.2 K and 4.2 K are summarised Table 4.11.

Table 4.11 – Theoretical total conductance at 1.2 K for CFRP and Vespel SP22 insulation

	CFRP	Vespel SP22
G_{01} at 1.2 K	64 $\mu\text{W/K}$	41 $\mu\text{W/K}$

These estimations of the conductance through the insulating elements give a good indication of the quality of the insulation of the heat switch base from the bracket that can be expected, despite not having accounted for the numerous TCRs present in the assembly.

Initially aiming for a G_{01} value of 24 $\mu\text{W/K}$ at 1.2 K, we note that even the less conductive Vespel SP22 insulating elements fall short of the target by roughly a factor 2. The TCRs are not accounted for in our calculation however. These will reduce the G_{01} conductance in our favour.

In Figure 4.25 the composite insulating elements are mounted on the heat switches. The Vespel SP22 set is mounted onto HS1 (without snubber) and the CFRP set onto HS2 (with snubber). We used a tightening torque of ~ 0.3 Nm to limit plastic deformation of the Vespel insulating elements (the nominal dry tightening torque for an M3 stainless steel screw is around 0.5 Nm).

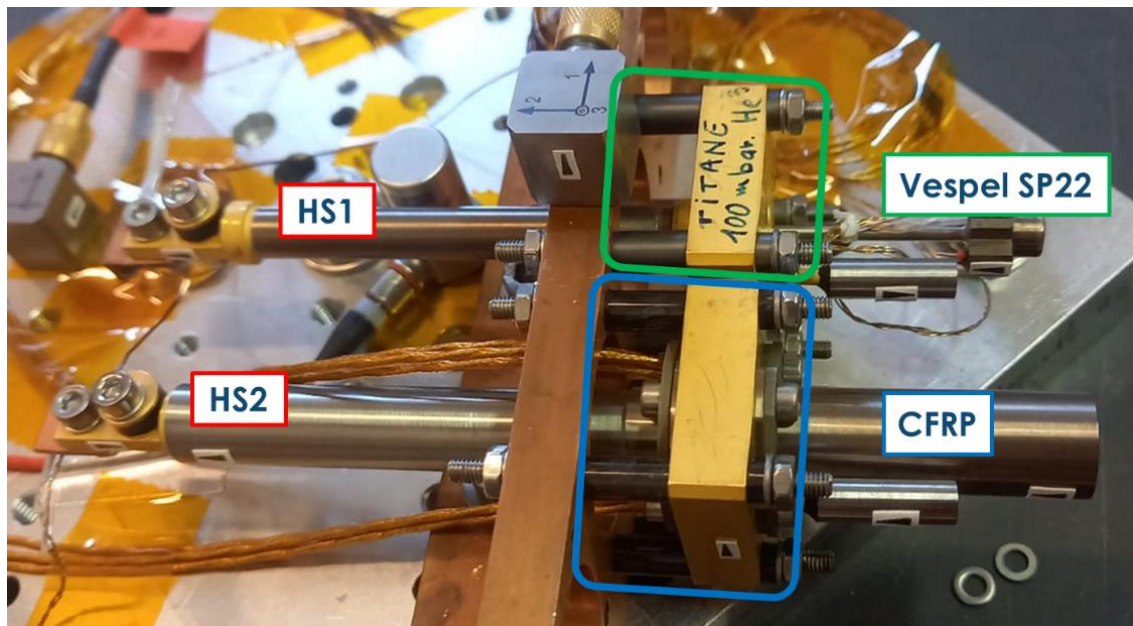


Figure 4.25 – HS1 and HS2 mounted with composite insulating elements (Vespel SP22 and CFRP)

The heat switches are mounted on a shaker here, shown in Figure 4.26. It is used for the mechanical characterisation presented in section 4.3.2. Before this, we will present a 3rd set of insulating elements.

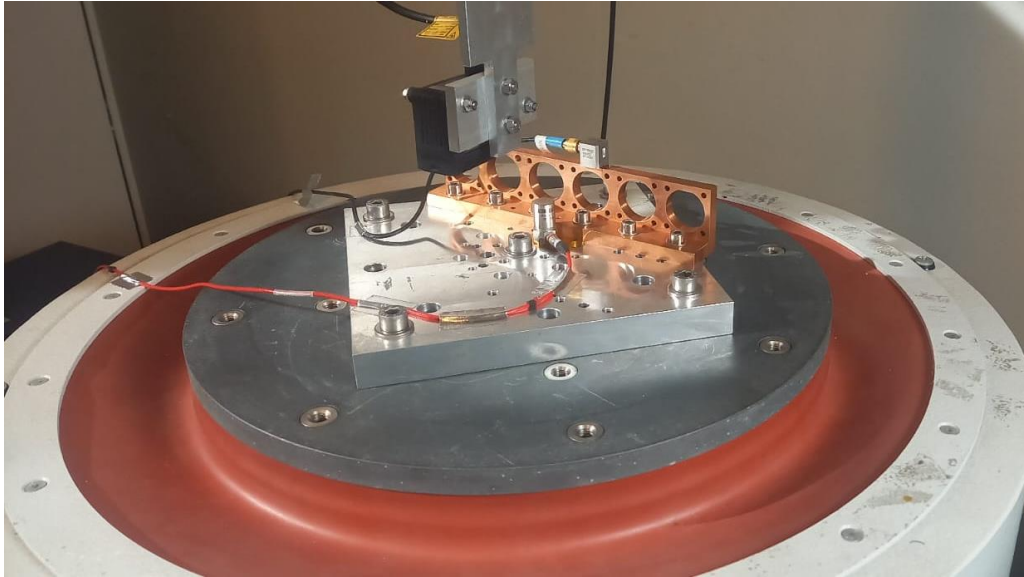


Figure 4.26 – Vibration shaker used for characterisation of heat switches

4.3.1.4 Steel washer insulation

As seen previously, at 1.2 K the conductivity of steel is around 5.27×10^{-2} W/m/K (see Table 4.10), whereas CFRP and Vespel SP22 are around 6 times and 22 times more insulating at this temperature (see Table 4.7). However, by maximising the number of contact surfaces (and so TCRs) through the stacking of many elements, stainless steel washers become a viable choice for our application.

In such a configuration, the TCRs are the dominant thermal resistance as steel has relatively high thermal conductivity. The TCR is very sensitive to the surface condition and effective contact area, influenced by the tightening torques. Unlike for the composite insulating elements, it is therefore difficult to estimate the total conductance between the bracket and heat switch base theoretically.

We therefore directly mounted 10 stainless steel washers head to tail on each screw between the bracket and base. Only HS2 was configured like so and it is shown in Figure 4.27. Here too we used a tightening torque of 0.3 Nm. This configuration was then directly characterised at 1.2 K for comparison with the composite material insulation. The characterisation of the 3 configurations is presented next.

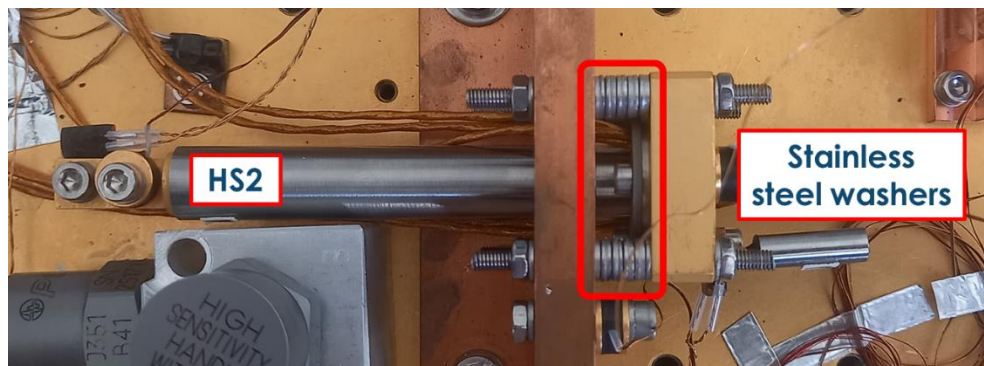


Figure 4.27 – HS2 mounted with stainless steel washers as insulating elements

4.3.2 Insulated base configuration characterisation

Before presenting the thermal characterisation of the heat switches mounted with the new insulating elements, we characterise the configurations mechanically.

4.3.2.1 Mechanical characterisation

To verify that the insulating elements have no impact on the mechanical resonance of the heat switches, specifically the 1st cantilever mode, we compare 4 configurations that we shall refer to as:

- “HS bare” (rigid base w/o instrumentation at tip)
- “HS instr” (rigid base w/ instrumentation at tip)
- “HS Vespel” (insulated base w/ instrumentation at tip and Vespel insulation)
- “HS CFRP” (insulated base w/ instrumentation at tip and CFRP insulation)

Both HS1 and HS2 are mounted in the first 3 configurations, whereas only HS2 is mounted in the CFRP configuration. The insulated base configuration using steel washers is not characterised mechanically, as it is assumed to be very stiff. A higher tightening torque can also be applied to ensure mechanical rigidity.

The above configurations are then mounted on a vibration shaker and the laser sensor is pointed to the heat switch tips. This is shown in Figure 4.28 (only HS1 mounted and in the Vespel configuration).

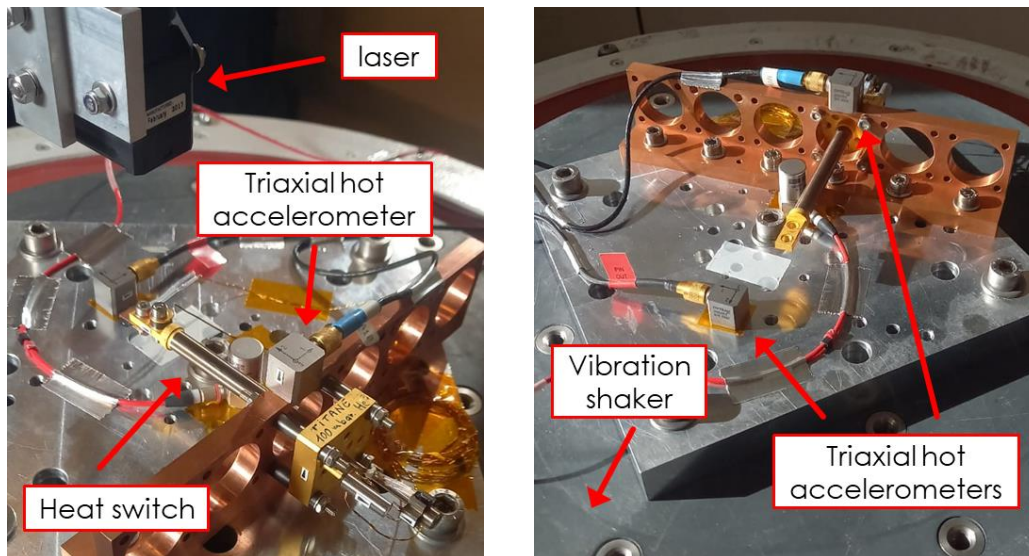


Figure 4.28 – Heat switch mounted onto vibration shaker (HS1 in Vespel configuration)

A logarithmic frequency sweep from 100 Hz to 350 Hz is carried out, at a speed of 0.5 oct/min and an excitation level of 0.15 g. Figure 4.29 shows the measured peak acceleration levels of the tips of HS1 (top) and HS2 (bottom) for each configuration tested, at 300 K.

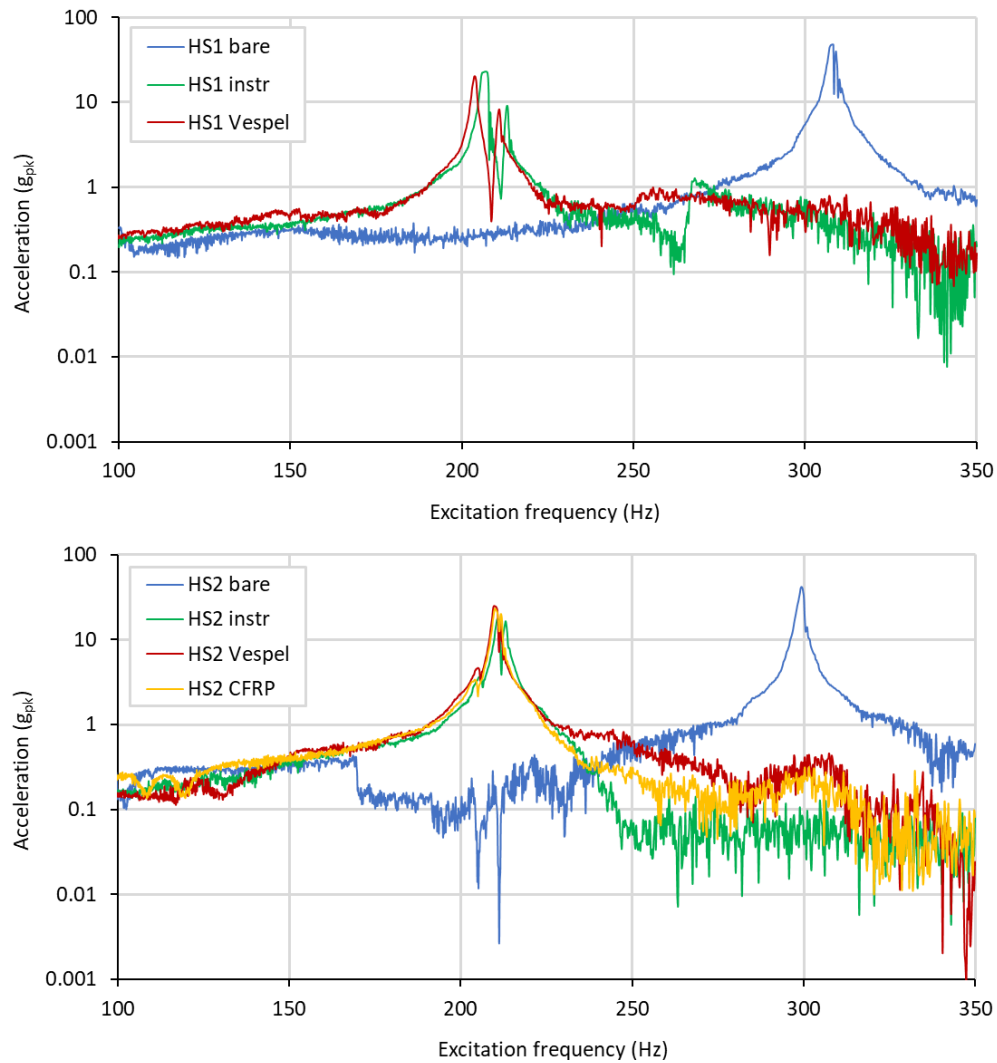


Figure 4.29 – Mechanical responses of tips of HS1 (top) and HS2 (bottom), for different heat switch configurations at 300 K

As expected, the observed resonance frequencies of both heat switches increase considerably when in the bare configuration (blue), without any instrumentation at their tips and so having a reduced mass. We also observe a seemingly double peak for “*HS1 instr*” and “*HS1 Vespel*” close to 210 Hz. Table 4.12 lists the observed resonance frequencies of each configuration. The Q factors are also shown, deduced from the amplitudes of each resonance and the known input acceleration level of 0.15 g.

Table 4.12 – Resonances and Q factors for different heat switch configurations

	Resonance		Q factor	
	HS1	HS2	HS1	HS2
<i>HS bare</i>	308.0 Hz	299.4 Hz	241	281
<i>HS instr</i>	207.0 Hz	210.9 Hz	152	119
<i>HS Vespel</i>	203.8 Hz	209.7 Hz	137	167
<i>HS CFRP</i>	-	210.1 Hz	-	154

Regarding the resonance frequencies of the “*HS instr*” configurations, we observe a slight shift compared to the resonances presented in section 2.2.4 that was done with the heat switch mounted on the cryostat cold plate at room temperature. HS1 shows a shift of about 7 Hz, whereas HS2 only shows a shift of 2 Hz.

Regarding the impact of the Vespel and CFRP insulating elements, we observe that the heat switch resonance frequencies have slightly reduced. This could be explained by reduced system stiffness, as the heat switch bases are now not rigidly mounted. Nevertheless, the frequency shifts are small enough to validate the mechanical behaviour of the heat switch in these insulated base configurations.

For HS1, we also observe what appears to be a double resonance near ~210 Hz. This could reflect an imperfection in HS1, whose unexpected mechanical and thermal behaviour is later observed too.

It is also interesting to observe the Q factors in Table 4.12. When bare, the average Q factor is 261, whereas when instrumentation and the insulating elements are added, it is 146. This is the first indication that the test setup directly affects the purity of the resonance of the heat switches and so by extension, its expected dissipation behaviour. We shall explore this in the subsequent sections of this chapter.

At this stage we must consider the possibility that the decrease in the Q factor could be due to a decrease in the resonance frequency. For a given acceleration level, displacement amplitude is inversely proportional to the square of the frequency. Hence, at a resonance frequency roughly 50% higher, displacement amplitudes are 56% lower. If some dissipative phenomena only appear at higher displacement levels, we can imagine higher Q factors when resonance frequencies are higher.

Thus we cannot yet discard the possibility that the Q factor change is simply due to the drop in resonance frequency (i.e. added mass) rather than the introduction of dissipative components (i.e. added damping).

Having now characterised the configurations considered most critical from a mechanical standpoint, (i.e. those using Vespel and CFRP), we now characterise the new configurations from a thermal standpoint.

4.3.2.2 Thermal characterisation

This was made possible by instrumenting the bases (in addition to the tips) with thermometers and heaters. When cooled, the conductances between the bracket and base G_{01} and between the base and tip G_{12} were measured through the regulation of the temperatures of the bracket T_0 , heat switch base T_1 and heat switch tip T_2 , as well as the measurement of the regulation powers at the base P_1 and tip P_2 .

We recall that it is assumed heat loads arriving on the heat switch are entirely evacuated through its base (i.e. no parallel heat fluxes through the Manganin instrumentation wiring of the base and tip), that there are no TCRs, that convection is absent and that radiation is negligible. The conductances are then:

$$G_{01} = \frac{P_1 + P_2}{T_1 - T_0} \quad (4.13)$$

$$G_{12} = \frac{P_2}{T_2 - T_1} \quad (4.14)$$

Material conductivities are temperature-dependant. The regulation temperatures chosen therefore impact the conductances of the set up. They were therefore measured for different combinations of temperatures T_0 , T_1 and T_2 , to evaluate the extent of this influence.

Table 4.13 shows the measured G_{01} and G_{12} values for both heat switches, mounted with the Vespel insulating elements and the steel washers, and for several combinations of regulation temperatures.

The CFRP configuration was not characterised thermally, as the Vespel configuration was found to be sufficiently rigid in the mechanical characterisation tests presented in section 4.3.2.1.

Table 4.13 – Measured conductances of test setup with Vespel and steel insulation

<i>Heat switch</i>	T_0	T_1	T_2	P_1	P_2	G_{01}	G_{12}	G_{01}/G_{12}
<i>HS1 Vespel</i>	4.220 K	9.000 K	12.000 K	2.36 mW	164 μ W	520 μ W/K	42 μ W/K	12.4
<i>HS1 Vespel</i>	4.220 K	5.000 K	7.000 K	190 μ W	50 μ W	297 μ W/K	21 μ W/K	14.1
<i>HS1 Vespel</i>	4.234 K	4.245 K	4.330 K	2.00 μ W	1.40 μ W	290 μ W/K	14 μ W/K	20.7
<i>HS1 Vespel</i>	1.251 K	1.301 K	1.601 K	1.55 μ W	1.26 μ W	56 μ W/K	4.2 μ W/K	13.3
<i>HS2 Vespel</i>	4.220 K	5.000 K	9.000 K	113 μ W	101 μ W	265 μ W/K	23 μ W/K	11.5
<i>HS2 Vespel</i>	4.266 K	4.290 K	4.360 K	3.00 μ W	1.40 μ W	183 μ W/K	20 μ W/K	9.1
<i>HS2 Vespel</i>	1.260 K	1.443 K	2.316 K	4.93 μ W	4.92 μ W	54 μ W/K	5.6 μ W/K	9.6
<i>HS2 steel</i>	1.234 K	1.294 K	1.559 K	9.14 μ W	1.09 μ W	171 μ W/K	4.1 μ W/K	42
<i>HS2 steel</i>	1.205 K	1.245 K	1.745 K	4.49 μ W	2.17 μ W	167 μ W/K	4.3 μ W/K	38

We recall that the target G_{01} conductance was 24 μ W/K at 1.2 K, reflecting a sensitivity to microvibration dissipation roughly one order of magnitude lower at the base than at the tip of the heat switch. Taking into account some constraints, the design value of the Vespel configuration was 41 μ W/K. For HS1 and HS2, the measured G_{01} conductances at 1.2 K are slightly higher (56 μ W/K and 54 μ W/K), which is not expected as the unaccounted for TCRs should have reduced the conductance. The values nevertheless remain acceptable for our dissipation measurements.

When the steel washers are mounted instead, the G_{01} conductance at 1.2 K increases to ~ 170 μ W/K (a factor ~ 3 increase compared to the Vespel configuration). Despite directly translating to a corresponding decrease in the sensitivity to microvibration dissipation, it is still acceptable for most of our measurements.

Considering both the mechanical and thermal behaviour of the different heat switch configurations, the tests presented for the remainder of this chapter use the Vespel and steel washer configuration. The microvibration dissipation behaviour observed with each of these 2 configurations will be compared.

Before presenting these though, we will first present other setup and test protocol improvements made.

4.3.3 Setup and test protocol

In addition to the new heat switch configurations, several other setup and test protocol improvements were made. The purpose is to improve the quality of our microvibration dissipation measurements.

Firstly, the measurement method used (see section 3.4) is adapted. We switch from the temperature slope method for dissipation measurement, to temperature regulation. In this way we measure dissipation continually while excitation frequency progresses and more precisely, as temperatures are constant and so thermal properties stable.

Next, the LabVIEW program is also adapted so that it carries out synchronous acceleration, temperature and electrical current (for temperature regulation) measurements. This means we are more confident about the true acceleration level that causes a given observed dissipation level, given that the mechanical response of the cryostat was found to vary considerably, as discussed in sections 3.1 and 3.2.

Regarding acceleration, it is now measured along the x , y and z directions as opposed to solely along z as done until now. In this way, we will be able to quantify the dissipative acceleration level more accurately, as we found that the transverse components were not always negligible (this environment characteristic was discussed in section 3.2.1). The sensitivity of microvibration dissipation to specific directions of mechanical excitation could then also be explored.

We also mounted additional accelerometers on the cold plate to eventually study all degrees of freedom (a preliminary exploration of rotational acceleration at the cold plate was presented in section 3.2). The test bench mounted with 6 accelerometers on the cold plate is shown Figure 4.30.

Lastly, finer post-processing techniques are also adopted such as band-pass filtering around the excitation frequency and comb filters at 50 Hz harmonics. This avoids the measurement of signals that are not directly induced by microvibrations at the excitation frequency being studied.

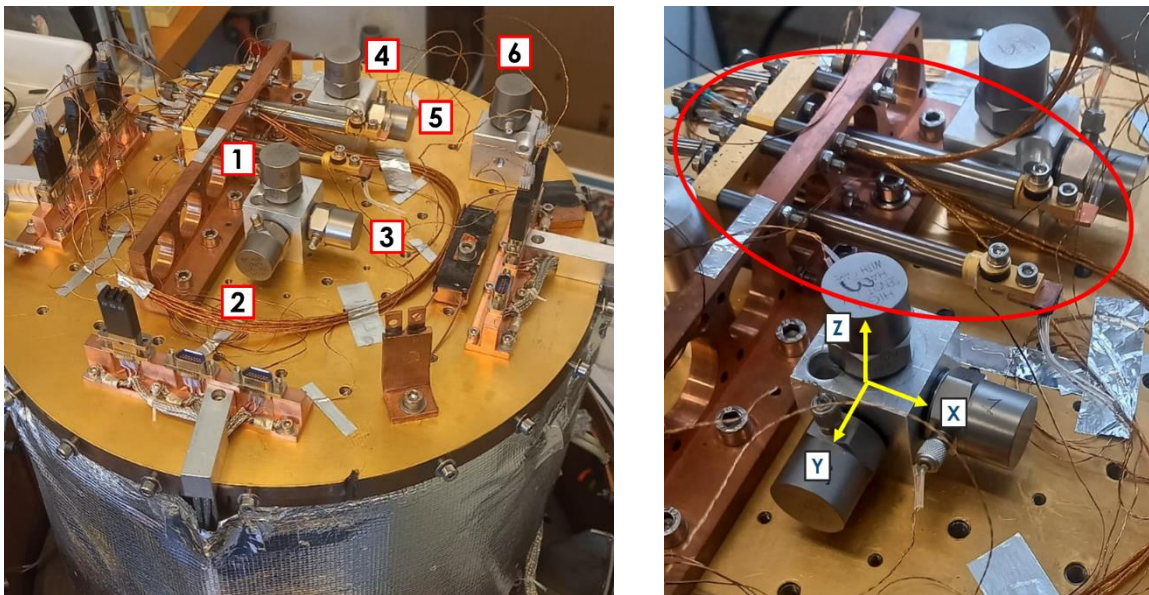


Figure 4.30 – Final setup of heat switch test bench with 6 cryogenic accelerometers and 2 insulated heat switches

4.3.4 Still high dissipation at tip

We can now present the first observations with the updated setup and test protocol, namely an insulated heat switch base and frequency sweeps with temperature regulation.

4.3.4.1 Regulation power drop

Given that both the base and tip of the heat switches are now insulated, we can regulate the temperatures of both with respect to the cold plate. As we are unaware of expected microvibration dissipation levels at this stage, we first choose relatively high regulation temperatures such that the regulation powers are high enough to not fall below zero due to the additional heat loads from microvibration dissipation. Indeed this would cause temperatures to rise and so the level of dissipation occurring to not properly be estimated.

For a cold plate temperature T_0 of 4.2 K, we set the temperatures of the base T_1 and the tip T_2 to 4.6 K and 7.2 K respectively. This corresponds to regulation powers at the base P_1 and at the tip P_2 of $\sim 40 \mu\text{W}$ and $\sim 70 \mu\text{W}$ respectively. Maintaining this temperature regulation and measuring the cold plate acceleration level synchronously, we carry out a frequency sweep from 200 Hz to 230 Hz (we expect the 1st cantilever mode of the heat switch to be in this range), at a relatively high speed of 50 s/Hz.

The resulting drop in regulation powers due to microvibration dissipation at both the tip (blue) and base (red) of HS1 is visible Figure 4.31. This is the first observation of microvibration dissipation using the temperature regulation and frequency sweep measurement methods. It is also the first dissipation measurement where we are *partially* informed on the dissipation location. At an excitation frequency near 215 Hz, we observe a drop of $40 \mu\text{W}$ of the tip regulation power and $8 \mu\text{W}$ of the base regulation power.

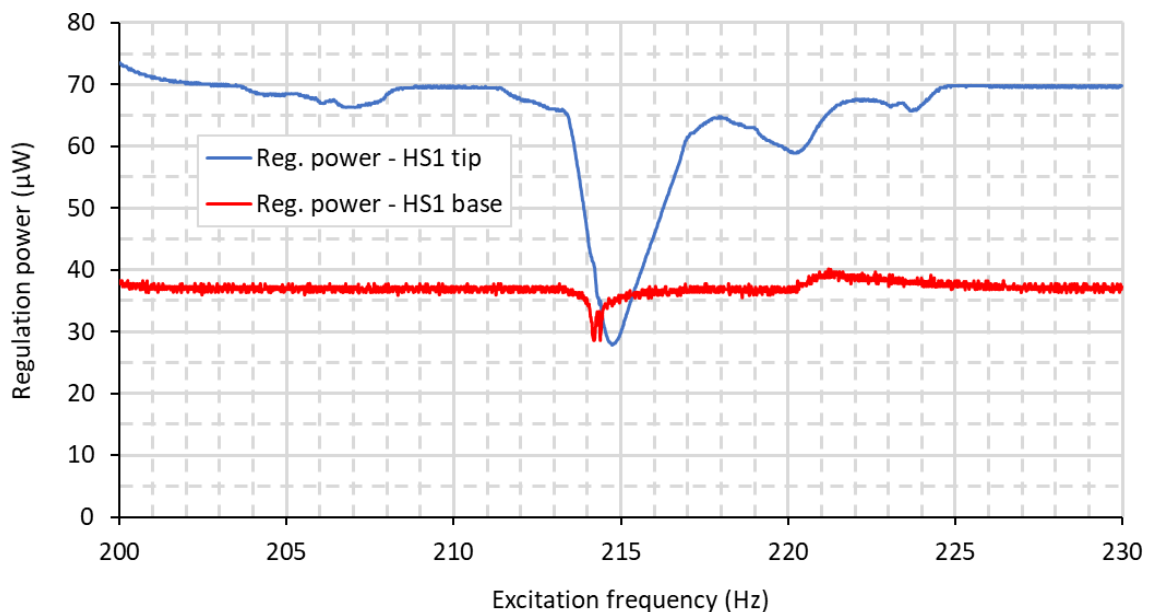


Figure 4.31 – Drop in regulation powers of HS1 due to microvibration dissipation, for a frequency sweep at 50 s/Hz and a cold plate temperature of 4.2 K

These drops in regulation power do not necessarily correspond to the level of microvibration dissipation occurring in the heat switch. They are rather an image of it, as we have previously seen for other load cases earlier in this chapter with the help of the thermal model. Applied here, we will now see how different dissipation cases may cause the same drops in regulation powers shown in Figure 4.31.

4.3.4.2 Analysis

Before insulating the heat switch base we had no knowledge of the total dissipation occurring in the heat switch. In the new configuration, the *increase* in the total heating received by the heat switch (regulation power plus microvibration dissipation) is given by the *drop* of the regulation power of the base of the heat switch ($8\ \mu\text{W}$ in our case). We also note that the regulation power of the heat switch tip falls by $40\ \mu\text{W}$. The total dissipation produced by microvibrations is therefore $48\ \mu\text{W}$. This dissipation may be spread along the heat switch in a profile or be concentrated at a point (or several points).

Thus the first load case (before microvibration dissipation) is $70\ \mu\text{W}$ of regulation power at the tip and the second load case (at peak microvibration dissipation) is $30\ \mu\text{W}$ of regulation power at the tip and $48\ \mu\text{W}$ of microvibration dissipation along the heat switch (a total of $78\ \mu\text{W}$).

Using the thermal model, we can show that if it is concentrated at a single point (i.e. the centroid position of the dissipation density profile, as discussed in section 4.2.2.3), there is only a single location where this $48\ \mu\text{W}$ of microvibration dissipation can be placed such that the regulation temperature T_2 is respected.

Figure 4.32 shows the 2 load cases (red crosses) and corresponding temperature profiles (blue) along the heat switch. We set T_1 to $4.6\ \text{K}$ and so do not show P_1 . Only the heat switch from the base to the tip is represented, not the insulating elements before the base.

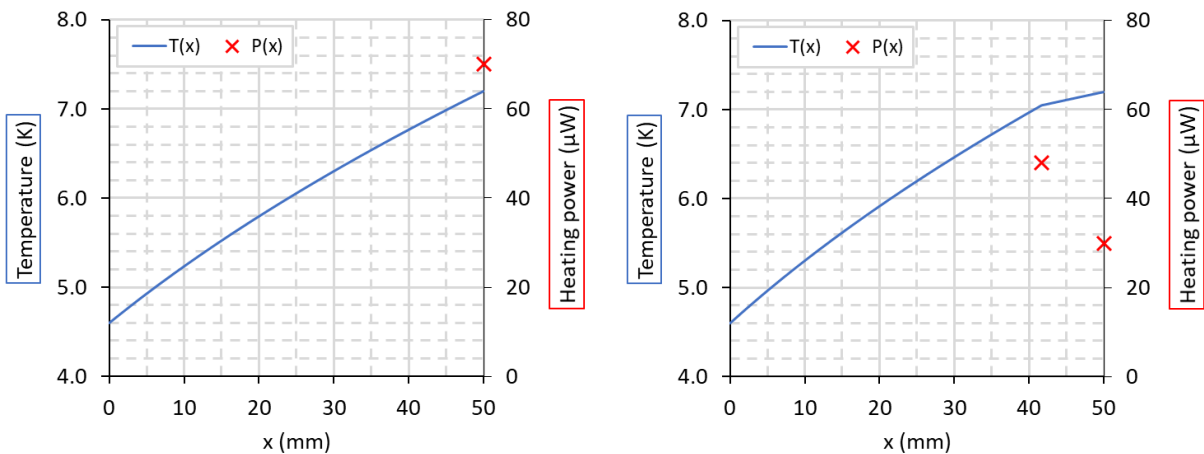


Figure 4.32 – Heat switch temperature profiles, before microvibration (left) and at peak microvibration dissipation (right)

For the first load case (left side of the figure), we observe that $70\ \mu\text{W}$ of regulation power at the tip predicts a tip temperature T_2 (at $x = 50\ \text{mm}$) of $7.2\ \text{K}$, as expected. For the second load case (right side of the figure), the regulation power at the tip falls to $30\ \mu\text{W}$ and the heat load of $48\ \mu\text{W}$ due to microvibration dissipation is placed at $x = 42\ \text{mm}$, namely about $10\ \text{mm}$ from the tip of HS1. This is the only location where it can be placed such that the tip temperature T_2 of $7.2\ \text{K}$ is respected. If this heat load was closer to the base of the heat switch T_2 would be lower and if it was closer to the tip T_2 would be higher.

As discussed in section 4.2.2.3, we expected microvibration dissipation to be greatest where mechanical stress was greatest, namely near the base of the heat switch, which does not appear to be the case. Furthermore, there is no reason to expect microvibration dissipation to be centred close to the tip of the heat switch ($\sim 10\ \text{mm}$) but not *at* the tip.

We can therefore reasonably assume that a small amount of microvibration dissipation is occurring at the base of the heat switch or close to it but that most is occurring at its tip. This can also be concluded from subsequent measurements. This is contrary to the expected location of dissipation. Insulating the base of the heat switch has highlighted that most of the total dissipation is in fact arriving at the tip of the heat switch, which we could not conclude from the preliminary measurements where the heat switch base was not insulated.

We imagine that different mechanisms could explain the microvibration dissipation seen at the heat switch tip. A plausible explanation is that dissipation is not occurring at the tip of the heat switch itself but rather in the wiring of the instrumentation there, as depicted in Figure 4.33. The oscillations of the tip of the heat switch are amplified due to its high Q factor, forcing motion of one end of the wiring attached to it. This in turn could cause the deformation of said wiring, damping oscillations and causing dissipation within them. This is a key turning point in the interpretation of microvibration dissipation observed during this thesis, having previously expected and sought it in the heat switch itself.



Figure 4.33 – Hypothesis: dissipation is occurring in wiring of heat switch tip instrumentation

Before presenting the advanced measurements that explore this, we will discuss 2 side effects that limited the dissipation range within which we were able to work.

4.3.5 Observed side effects

The first unwanted side effect appears for too high mechanical excitation levels. The second involves threshold and saturation effects. Both limit the range within which we were able to work during our tests.

4.3.5.1 Thermal shorting of heat switch

The microvibration dissipation tests of this thesis have involved mechanical excitation levels below $40 \text{ mg}_{\text{rms}}$, with a focus at levels below $10 \text{ mg}_{\text{rms}}$. A side effect we observed at excitation levels exceeding $40 \text{ mg}_{\text{rms}}$ is what appears to be thermal shorting between the base and tip of the heat switch, which effectively thwarts any type of dissipation measurement at the tip.

Figure 4.34 shows a case of temperature regulation at 5 K (base) and 9 K (tip) for HS1. The cold plate is at 4.2 K, and we performed a sweep from 210 Hz to 220 Hz at a high shaker drive voltage of $500 \text{ mV}_{\text{pp}}$.

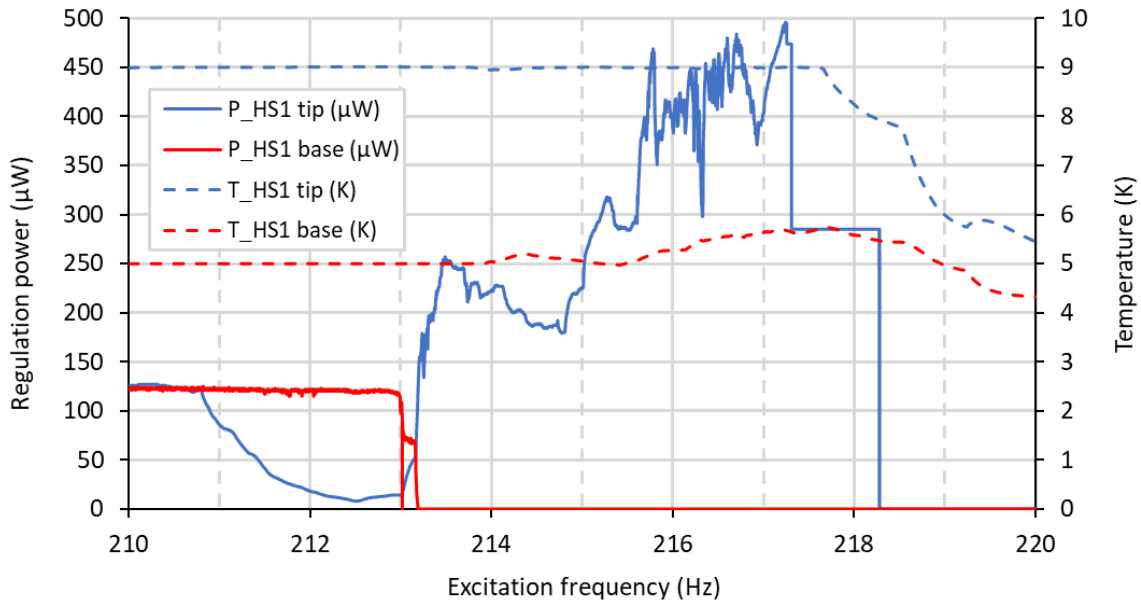


Figure 4.34 – Case of thermal shorting between base and tip of heat switch (HS1)

At first, temperatures and regulation powers are stable. We observed no microvibration dissipation. Just before 211 Hz, microvibration dissipation appears at the tip of the heat switch, causing the regulation power to drop there. The base regulation power remains stable until 213 Hz, when it falls to zero, suggesting that a large heat load suddenly arrives there. The regulation power at the tip also rises rapidly to maintain its temperature set point of 9 K. This then suggests a large heat load *leak* from the tip of the heat switch. We thus suspect a thermal short between the base and tip.

From 213 Hz onwards this thermal shorting continues, as the tip regulation power continues to rise sporadically and the base regulation power remains at zero. We observe that from 214 Hz onwards, the total heat load arriving at the base exceeds the power required to maintain it at 5 K, causing it to drift.

We theorise that mechanical contact between the copper base and tip (normally separated by the 100 μm gap when not under strain) causes this thermal shorting, due to the high mechanical excitation levels. Another possibility is that localised heating (say from microvibration dissipation) is causing the helium gas to desorb into the gap, thus causing high conductance between the base and tip. This thermal shorting effect was observed for HS1 only, though may appear for HS2 too at higher excitation levels. We did not test HS2 to very high levels to preserve its thermal behaviour for our microvibration dissipation tests.

As mentioned when presenting the heat switches in section 2.2, the majority of the results presented in this thesis have focused on measurements made with HS2. The thermal shorting observed with HS1 that is presented here is one of the motivations for doing so.

4.3.5.2 Threshold and saturation effects

As discussed, most tests presented in this chapter have focused on mechanical excitation below 10 mg_{rms} . Below this level, the thermal shorting effect just discussed did not appear. However, some of our dissipation measurements seem to indicate the presence of threshold and saturation effects.

Indeed, there appears to be a dissipation *threshold* (no dissipation for a non-zero excitation level) when observing peak dissipation versus acceleration. We find that the highest acceleration level that causes no dissipation depends on the test configuration, varying between 0.5 mg_{rms} and 2.5 mg_{rms}.

There also appears to be dissipation *saturation* in some cases. For a higher Q factor – causing higher peak dissipation for a given acceleration level – saturation appears for excitation as low as 4.5 mg_{rms}.

If indeed both threshold and saturation effects are present, the proportionality between dissipation and the square of the acceleration level (as predicted by the dissipation model), may only be true within certain bounds. In addition, these bounds were found to vary depending on the test configuration we used.

4.3.5.3 Discussion side effects

To avoid thermal shorting and irreversible damage to the heat switch we therefore chose to study microvibration at mechanical excitation levels below 10 mg_{rms}. Saturation effects that seem to appear for specific test configurations further back this choice. Regarding a lower limit of mechanical excitation and dissipation, we cannot reliably study the region much below 1 mg_{rms}. These bounds fall within the ranges we are capable of measuring with our detection chain, as discussed in the conclusion of chapter 2.

4.3.6 Summary

To conclude this section, it is worth summarising the key takeaways.

Having justified the need for insulating the heat switch, we machined and characterised several sets of insulating elements, both mechanically and thermally. The configurations tested were found to have little effect on the 1st cantilever mode resonance frequency. We expect the impact of the steel washer configuration to be even lower, given its higher rigidity. This is why it was not characterised mechanically.

For thermal characterisation, we achieved a conductance ratio G_{01}/G_{12} close to the target of 10, in the Vespel configuration and for various regulation temperatures below 12 K. In the steel washer configuration, the average conductance ratio was 40 (see Table 4.13). This is still an acceptable level of decoupling between the base and the bracket. The advantage of this configuration is that mounting is easy and repeatable, and the insulating components (standard steel washers) are easily at hand.

We also made other setup and test protocol improvements. In particular, to address the measurement difficulties highlighted in section 4.1.8, mechanical and thermal measurements are now synchronised and the acceleration vector is measured along x, y and z, as opposed to only along z. Regarding the measurement method, we switch to temperature regulation as opposed to the temperature slope method.

Finally, we presented the first observations made with an advanced test configuration. We showed drops in the regulation powers at the base and tip of the heat switch, due to microvibration dissipation. Contrary to our expectation, we found a much larger dissipation level at the tip of the heat switch than at the base.

Our hypothesis is that microvibration dissipation is not occurring in the heat switch tip itself but rather elsewhere, such as in the instrumentation or its associated wiring. This interpretation is the first in which

microvibration dissipation other than in the heat switch itself is considered. As we shall see next, this also applies to microvibration dissipation that appears at the base of the heat switch.

We will now study the impact of wiring and of the different test configurations on microvibration dissipation, in an attempt to identify the sources and locations of this dissipation in our test setup.

4.4 Advanced measurements

In this thesis in general, we have dealt with systems showing relatively high Q factors, from the standpoint that the bandwidth of microvibration dissipation was very low. This means we must take care to use slow enough sweep speeds when performing sweeps. To accurately deduce dissipation, sweep speeds as slow as 1 hour/Hz are sometimes needed. This means that if the dissipative frequency band is not first identified, sweeping across even a 10 Hz bandwidth would be tedious. We therefore begin by performing high bandwidth sweeps, at a high speed and at a high excitation level, such as in Figure 4.31 where we sweep at 50 s/Hz. We then reduce the excitation bandwidth, sweep speed and excitation level.

For these finer tests, we carry out synchronous 3-axis acceleration and dissipation measurements, at fixed of shaker drive voltages. The shaker drive voltages have no meaning to us in themselves. We adjust them aiming for acceleration levels of the order of 100 μg to 10 mg and dissipation levels of the order of 100 nW to 10 μW . In chapter 2 we validated the capabilities of our detection chain in these ranges.

In this section, we will study the impact on microvibration dissipation, of wiring and of the test configuration. We will then explore the correlation of the model with the various dissipation measurements.

4.4.1 Impact of wiring

Using the laser sensor, we are able to study the Q factor of the heat switches mechanically at room temperature when the heat switch is mounted on the cold plate (cryostat open). We point the laser to the tips and perform frequency sweeps around the 1st mode resonance and perform several runs of the same test to verify repeatability. The result is shown in Figure 4.35, the heat switch in the Vespel configuration.

From these displacement curves we deduce the Q factor from the bandwidth definition, as per equation (1.48). Here it averages 140. The small superimposed oscillations ~ 0.1 Hz wide are a measurement artefact. They likely correspond to the resolution bandwidth of the spectrum analyser (which is a function of the window length) or could be due to windowing effects that cause spectral leakage.

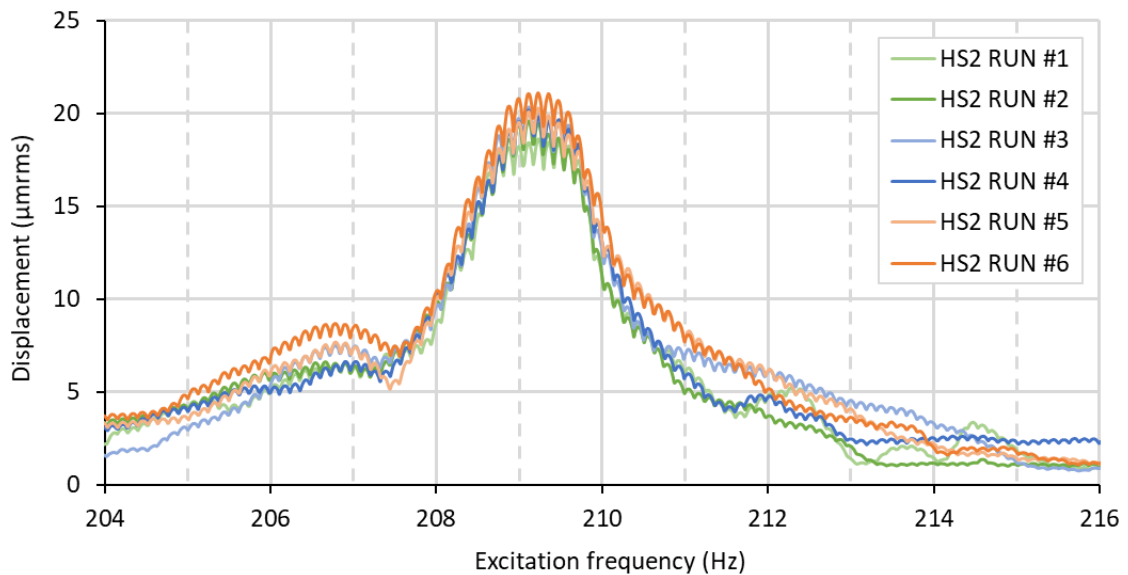


Figure 4.35 – Displacement measurements of HS2 at room temperature (in Vespel configuration)

From room temperature measurements (laser displacement sensor), the Q factor calculation is more precise as vibrations are induced directly at cold plate level (rather than from the outside of the cryostat when cold and closed). This means the injected acceleration profile is very flat. The displacement curves of the heat switch tip are therefore not distorted, allowing more precise Q factor calculations.

Carrying out tests at ambient temperature is easy and can provide insights into the microvibration dissipation behaviour that we may expect. The next subsections explore different test setup adjustments.

4.4.1.1 No instrumentation

We first remove all the instrumentation and its associated wiring from the tip of the heat switch. As we reduce the mass, we find that the resonance frequency has increased: from 210 Hz to roughly 297 Hz. Again, we carry out several runs and calculate the Q factors from the displacement curves. The impact of removing the instrumentation of HS2 on the Q factor is shown in Figure 4.36.

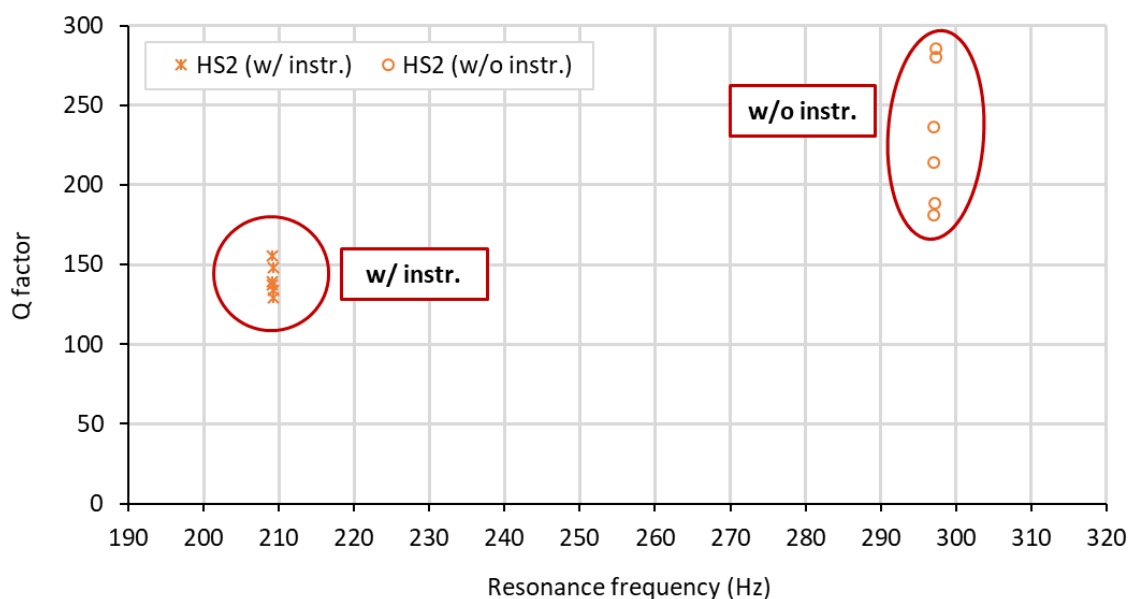


Figure 4.36 – Impact of removal of instrumentation on Q factor of HS2 at room temperature (in Vespel configuration)

The Q factor has increased from ~ 140 to ~ 230 . This suggests that we have altered the dissipative behaviour of our setup. However as discussed in section 4.3.2.1, the increase in the Q factor could be due to an increase in the resonance frequency. If some dissipative phenomena only appear at higher displacement levels, we can imagine higher Q factors when resonance frequencies are higher resonance.

To eliminate this possibility, we introduce a dummy mass to reduce the resonance. By trial and error a copper block is sanded down until the resonance frequency is lowered back to ~ 210 Hz. Finally, the Q factors are calculated again with the dummy mass attached to the tip. This procedure of calculating the Q factor first with instrumentation, then without instrumentation and lastly with a dummy mass, is carried out for both heat switches. Figure 4.37 shows HS2 in each setup: with instrumentation (left), without instrumentation (middle) and with a dummy mass (right).

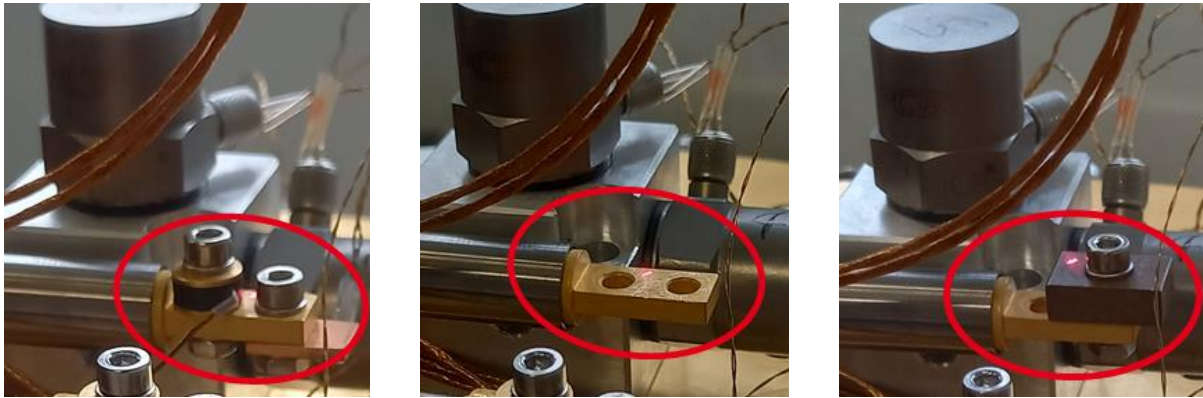


Figure 4.37 – HS1 with instrumentation (left), without instrumentation (middle) and with dummy mass (right)

The resulting calculated Q factors for each setup and each heat switch are summarised in Figure 4.38.

We observe that at each evolution of the setup of the heat switch tip (from left to right in Figure 4.37), the average Q factor increases. In particular, when we compare the setups at similar resonance frequencies close to 200 Hz, replacing the instrumentation and wiring with a dummy mass has increased the Q factor by a factor 2.5. This suggests that the instrumentation and its wiring strongly impact the mechanical dissipation behaviour of the heat switches.

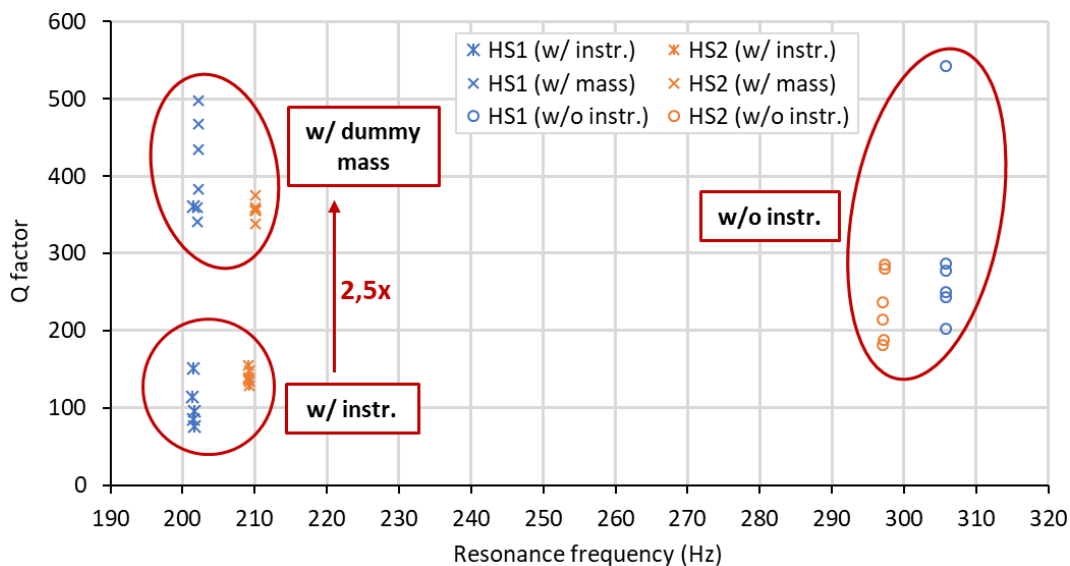


Figure 4.38 – Impact of different heat switch tip setups on the measured Q factor of HS1 and HS2 (in Vespel configuration)

We also note that the spread of calculated Q factors is high for HS1 (blue). As discussed in section 4.3.5 on some observed side effects, at high excitation levels we believe to have observed thermal contact between the base and tip of HS1, making dissipation measurements impossible. Even when this effect was absent, HS1 showed abnormal microvibration dissipation behaviour. We therefore focused on HS2.

To see how the change in Q factor at room temperature influences the mechanical dissipation behaviour of the heat switch, we must nevertheless instrument the tip of the heat switch. We re-instrumented HS2 while paying special attention to how the wiring was routed in an attempt to maintain a high Q factor.

4.4.1.2 Wire rerouting and Q factor

Prior to removing the instrumentation from the heat switches, all wiring from instrumentation was loosely spun in a spiral and thermalized to the cold plate. We now decide to route the 4-wire Manganin leads of both the thermometer and heater at the tip of the heat switch parallel to the titanium tube, and thermalize them to the bracket with aluminium tape. This is shown in Figure 4.39.

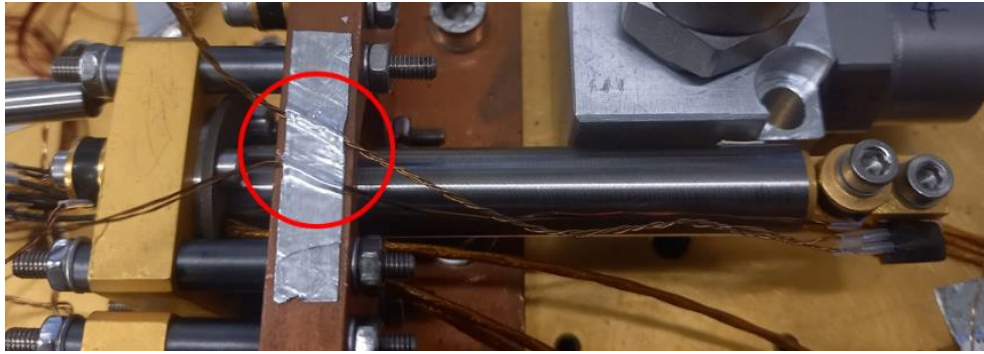


Figure 4.39 – Rerouting of heat switch tip wiring along its length

We then carry out new displacement measurements and calculate the Q factor. The values for the setups with the old routing, dummy mass and new routing are summarised in Figure 4.40. Here we only test HS2.

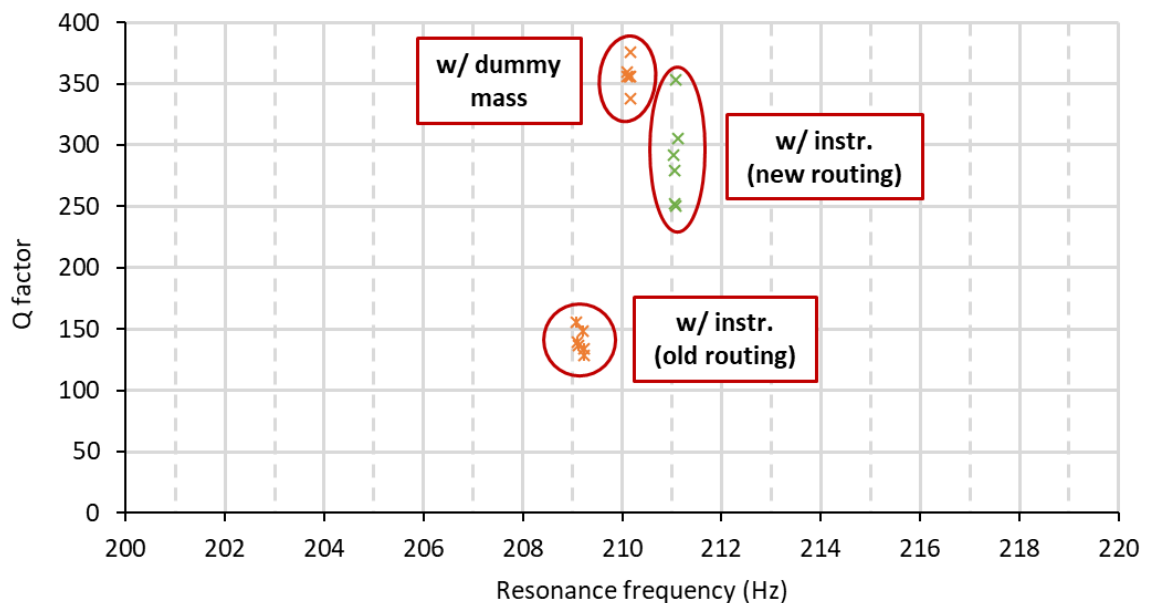


Figure 4.40 – Impact of heat switch tip wire rerouting on the measured Q factor of HS2

We find that our method of rerouting has not reduced the Q factor (~ 290) much, compared to the dummy mass setup (~ 360). With the old routing, the average Q factor was 140. Simply routing the wiring differently is found to significantly affect the damping of our system. We expect this to have a high impact on measured microvibration dissipation levels too, which we will now study.

4.4.1.3 Wire rerouting and dissipation

In the case *before* wire rerouting, shaker drive voltages between 30 mV_{rms} and 200 mV_{rms} were found to correspond to the target acceleration and dissipation levels. Broad frequency sweeps identified dissipation in the 219-221 Hz range for HS2. The dissipation levels at the base and tip of HS2 at 1.2 K are shown in Figure 4.41. They are obtained by zeroing the drops in regulation powers, such as those shown in Figure 4.31. Having assumed microvibration dissipation only at the base and tip, the y-axis of these plots therefore directly corresponds to dissipation power. The corresponding acceleration curves were presented in section 3.5.1, in the top part of Figure 3.17, ranging between 1 mg_{rms} and 5 mg_{rms}.

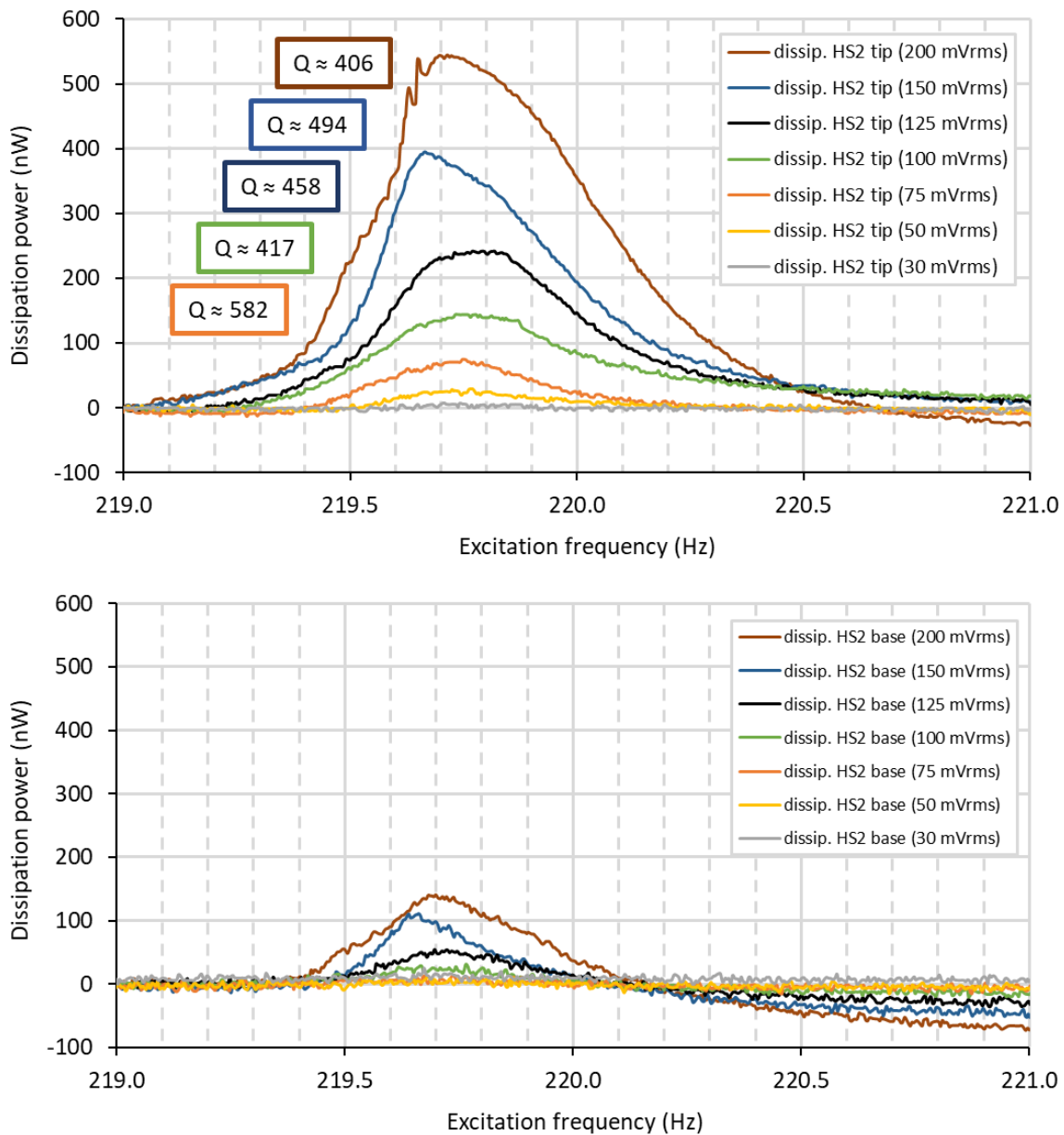


Figure 4.41 – Microvibration dissipation levels at tip (top) and base (bottom) of HS2 (before wire rerouting) at 1.2 K

We can make several observations from these dissipation curves. First, we still see most of the dissipation at the tip of the heat switch rather than at its base: for the highest excitation level (200 mV_{rms}) we measure 540 nW at the tip and only 140 nW at the base.

Next, we observe a higher Q factor when our test setup is cooled, from ~ 140 at 300 K to ~ 470 at 1.2 K. The excitation levels caused too little dissipation at the heat switch base to reliably deduce a Q factor there, as well as the lowest excitation level (30 mV_{rms}) for the tip. We recall that the dissipation curves are slightly distorted here due to the injected acceleration level, as discussed in section 3.5.2. For apparent Q factors below 4000 the error in its calculation is less than 20%, which is still acceptable for our analysis.

We recall that deduced from the dissipation curves here, these are *apparent* Q factors. As discussed in chapter 3, the mechanical feedback from the heat switch introduces an error in the calculation of the Q factor from the dissipation curves. This error becomes large when the apparent Q factor is very high (>4000). For lower Q values the error was estimated to be within 20% of the true value. Thus here, where we deduce values below 600, the Q factor precision remains acceptable and does not hinder the analysis of our results, as we are only interested in orders of magnitude.

Comparing these dissipation curves (ranging between 100 nW and 500 nW) and the acceleration curves (ranging between 1 mg_{rms} and 5 mg_{rms}) presented in Figure 3.17 of section 3.5.1, we note roughly 500 nW of microvibration dissipation for 5 mg_{rms} of mechanical excitation, *before* wire rerouting.

We also measured dissipation levels *after* having rerouted the wiring at the tip of the heat switch. Shaker drive voltages were adjusted such that the injected acceleration levels, before and after wire rerouting, were roughly equal. This corresponded to shaker drive voltages between 15 mV_{rms} and 50 mV_{rms}. Broad frequency sweeps now identified dissipation in the 220-222 Hz range for HS2. The new microvibration dissipation curves at 1.2 K, after having rerouted the wiring, are shown in Figure 4.42. The acceleration levels that cause them were presented in section 3.5.1 too, in the middle of Figure 3.17.

The first major observation, comparing these curves to the dissipation curves before wire rerouting (see Figure 4.41), is that peak dissipation is now much higher despite similar acceleration levels, up from 500 nW to over 3.5 μ W. This corresponds to roughly 3.5 μ W of microvibration dissipation for 4 mg_{rms} of mechanical excitation (up from roughly 500 nW for 5 mg_{rms} *before* wire rerouting).

The higher dissipation levels can be explained by a Q factor that is now significantly higher too. At similar dissipation levels, say around 500 nW, rerouting has increased the apparent Q factor measured at 1.2 K from around 400 to 2400. As for the case before wire rerouting, here too the dissipation curves are distorted. The error in its calculation is less than 20% here, which is still acceptable for our analysis.

Another observation is that the Q factor seems higher at lower excitation levels. This could be explained by dissipative phenomena that are important at higher excitation levels (or that are negligible at lower ones). As the range of dissipation levels is larger here, this effect may be more visible than with dissipation observed before wire rerouting. Lastly, we also note that dissipation is still mostly observed at the tip.

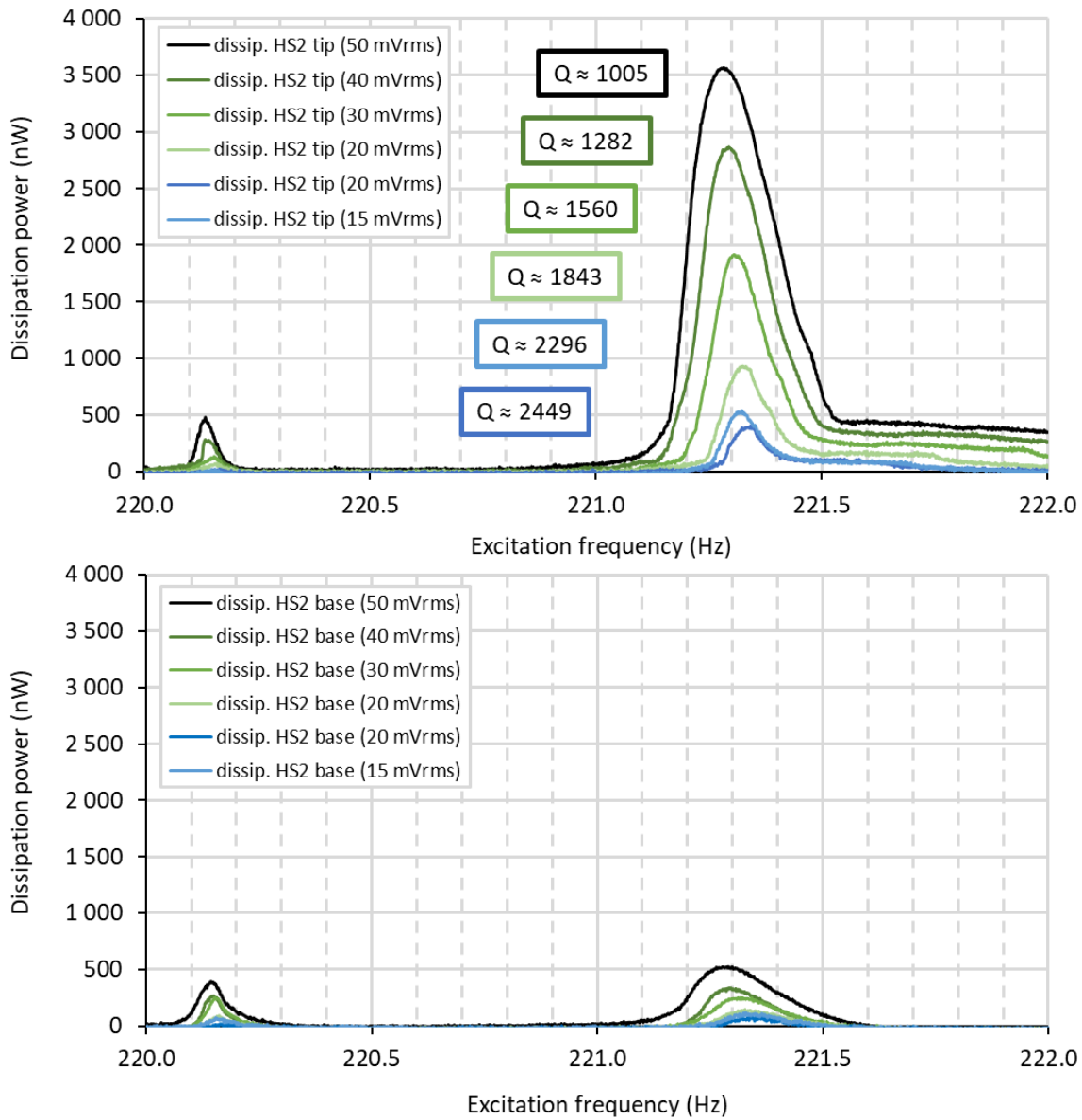


Figure 4.42 – Microvibration dissipation levels at tip (top) and base (bottom) of HS2 (after wire rerouting) at 1.2 K

4.4.1.4 Analysis

After rerouting the wiring at the tip and despite similar acceleration levels (see Figure 3.17), microvibration dissipation at “resonance” is much higher at both the tip and base of the heat switch.

The greatest impact is at the tip, where peak dissipation increases from 550 nW to 3.5 μ W. The bandwidth of dissipation is now much lower too, the Q factor increasing from 400 to 2400 after rerouting (at equal dissipation). We conclude that by rerouting the wiring we have *reduced* damping (a higher Q factor). Though counterintuitive, this in turn *increased* peak dissipation (at resonance) and caused a sharper *decline* in dissipation as we move away from the resonance.

Our model also predicts proportionality between peak dissipation and the Q factor, as per equation (1.45). Here dissipation increases by a factor 6.4 and the Q factor by a factor 6.0, which agrees with the model.

Dissipation at the base increased too, despite only rerouting the wires at the tip. For now, our hypothesis is that greater displacement of the tip due a higher Q factor increases the stress at the base of the heat switch. This could explain why dissipation at the base increases from roughly 150 nW to 500 nW.

Comparing dissipation to acceleration, *before* wire rerouting we measure around 500 nW of dissipation for 5 mg_{rms} of mechanical excitation at the tip, whereas *after* rerouting this figure is closer to 3.5 μW for 4 mg_{rms} . We have thus increased microvibration dissipation in our test setup by up to 1 order of magnitude, simply by rerouting wiring of our instrumentation. As the heat switch itself has not been modified between these tests, this confirms the hypothesis of section 4.3.6 that microvibration dissipation mostly occurs elsewhere than in the heat switch itself. We conclude that forced oscillations of wiring attached at one end to the heat switch significantly impact microvibration dissipation levels.

With the preliminary measurements of section 4.1, we measured Q factors between 1000 and 1500, and dissipation levels around 2 μW for 4 mg of mechanical excitation (with the caveat that dissipation occurs at the heat switch tip, which we now know not to be true). These values fall between the 2 cases presented here, where not only the wire routing has changed (its impact on dissipation now confirmed to be high) but also the test configuration. We mounted the base rigidly for the preliminary tests but used the Vespel configuration here. This suggests the test configuration used also impacts microvibration dissipation.

A possible explanation is that dissipation occurs in the insulating elements themselves. Another is that due to differential contraction between components of the bracket-base assembly upon cooling, the mechanical transmission from the cold plate to the insulated heat switch base may have changed. This aspect was overlooked in the tests discussed here with the first insulated base configuration using Vespel, though may explain differences observed between it and other configurations that we will now present.

4.4.2 Impact of configuration

We will now study microvibration dissipation with the steel washer configuration (it was presented in section 4.3.1). A different test setup provides other insights into the sources of microvibration dissipation.

4.4.2.1 More dissipation at base

For each of the 4 screws, we mounted 10 steel washers head to tail between the base of the heat switch and the bracket (see Figure 4.27). The optimal wire routing that identified with the Vespel configuration is kept here too. The new dissipation curves at 1.2 K for the base and tip of HS2 are visible Figure 4.43.

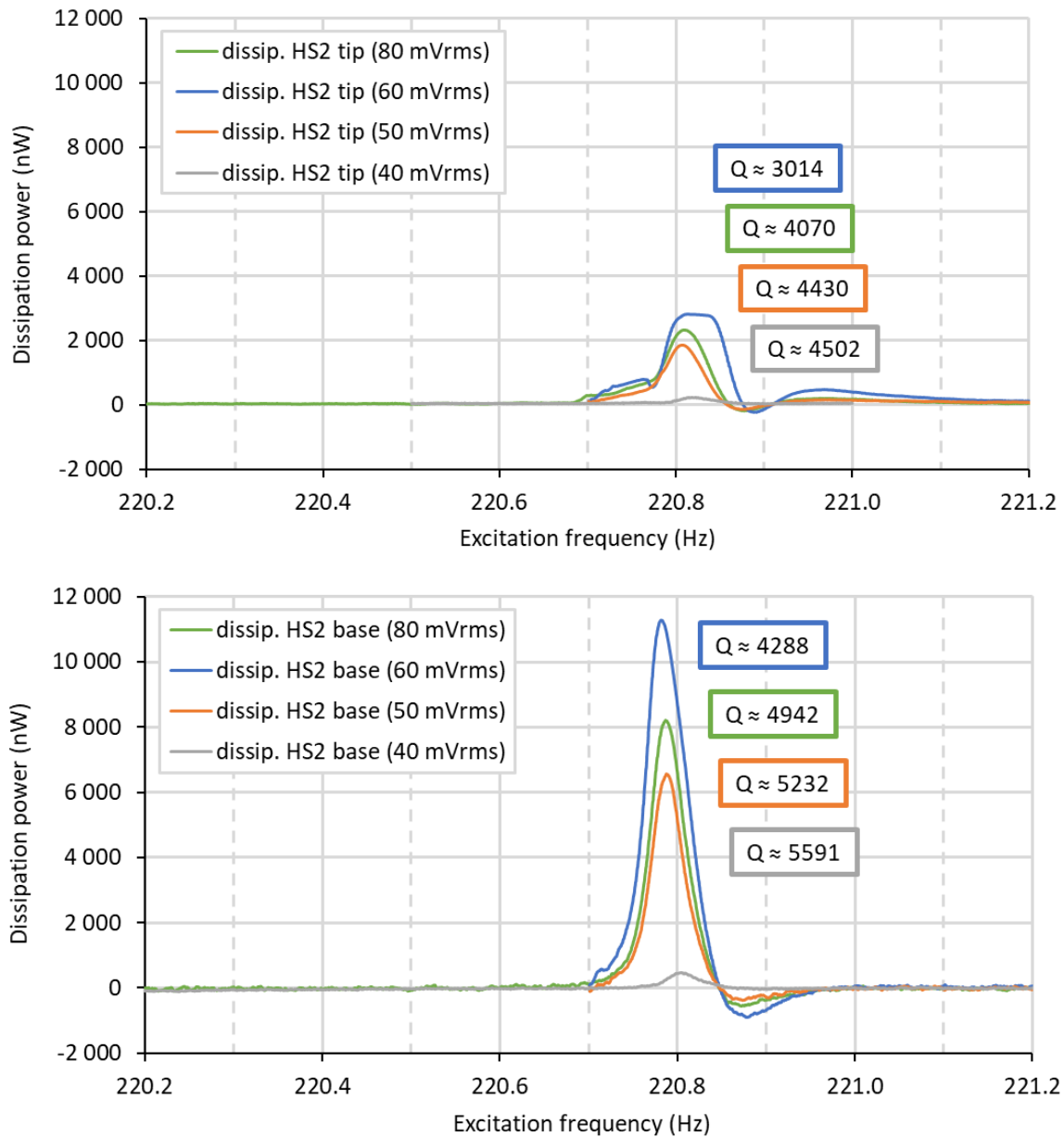


Figure 4.43 – Microvibration dissipation levels at tip (top) and base (bottom) of HS2 in steel washer configuration at 1.2 K

We can make 2 important observations here. The first is that the apparent Q factors are now much higher, of the order of 4000 to 5000 (versus 1000 to 2000 in the Vespel configuration after rerouting). The second is that dissipation now appears to be higher at the base of the heat switch ($\sim 11 \mu\text{W}$) than at the tip ($\sim 3 \mu\text{W}$). This is the first instance of microvibration dissipation being greater at the base of the heat switch.

Here too, we present the corresponding transverse acceleration levels (perpendicular to the heat switch axis) at the cold plate in section 3.5.1, in the bottom part of Figure 3.17.

4.4.2.2 Analysis

Having switched from the Vespel to the steel washer insulation, we observe very different microvibration dissipation behaviour. The steel washer configuration is the first (and only) with which more dissipation is observed at the base ($\sim 11 \mu\text{W}$) than at the tip ($\sim 3 \mu\text{W}$). We recall that the high-Q wire routing configuration

deduced in section 4.4.1.3 is kept here. As suspected, we can therefore confirm that the manner in which our test object is mounted significantly affects the sources and locations of microvibration dissipation.

This could be because the Vespel insulating elements display dissipative behaviour and so removing them would increase the purity of the resonance (a higher Q factor) of our system. Another explanation is that mechanical transmission to the heat switch base (and so mechanical amplification) was degraded by the Vespel elements due to a lack of tightening. Indeed when cooled, the differential contraction between Vespel and stainless steel may have caused a loosening of the screws fixing the heat switch base to the bracket (which could have been addressed using Belleville washers). Were the heat switch less rigidly mounted, bending stress at the base would be lower. It could still oscillate as a rigid body however, allowing oscillations of its tip and thus deformation of the wiring there. This could explain why barely no dissipation was observed at the base of the heat switch, whereas a significant amount was seen at the tip due to – as concluded earlier in this chapter – wire deformation.

If dissipation indeed occurs in the insulating elements (e.g. due to rubbing between the steel washers) rather than at the base itself, we can use the same reasoning as for dissipation observed at the heat switch tip (see section 4.4.1). The drop in regulation power at the base may not correspond to the actual dissipation occurring there. It could simply be a thermal side effect of dissipation occurring in the washers.

We also observe much steeper injected acceleration peaks (see section 3.5.1), with up to a factor 5 variation in amplitudes. This is explained by the higher Q factors of the heat switch that influence the acceleration level at the cold plate due to mechanical feedback, as discussed in section 3.5.1. The higher the Q factor, the greater we expect this mechanical feedback to be. As discussed in section 3.5.2, large variations in the acceleration level introduce imprecision in the calculation of the Q factor from the bandwidth definition. Thus the larger the Q factor, the less precise our calculation of it is from dissipation curves. Our deduced values of exceeding 5000 are thus likely overestimated. For the correlation of these dissipation curves with the model in section 4.4.3.3, we therefore adopt a more conservative value of 2000 instead. The earlier conclusion that the Q factor has increased here, is nevertheless still valid.

We will now explore the correlation between the hitherto presented dissipation curves and the dissipation model presented in chapter 1. This will shed new light onto the measurements presented so far.

4.4.3 Correlation with model

As presented in section 4.1.7 regarding the preliminary dissipation measurements, it is worth exploring the correlation between the dissipation measurements presented in section 4.4.1.3 and the model. We will first study peak dissipation (i.e. at a single excitation frequency), before further studying microvibration dissipation levels observed across the full excitation bandwidth.

4.4.3.1 At peak dissipation

From the dissipation and acceleration curves presented in section 4.4.1.3, we can plot the maximum microvibration dissipation as a function of acceleration. Figure 4.44 shows this for both the base and tip of HS2, both before and after wire rerouting in the Vespel configuration.

The dissipation model outlined in chapter 1 predicts dissipation that is proportional to the square of the injected acceleration level, as per equation (1.45). Fitting power laws to the 4 data sets of Figure 4.44, we observe an exponent value averaging 2.3, which is in agreement with the model prediction. We should note that these fits exclude the last data point for each of the post-rerouting cases (blue) at $\sim 4.4 \text{ mg}_{\text{rms}}$, as these appear to show saturation of the dissipation level. The power law fits also all pass through the origin, thus assume no threshold effects. Threshold and saturation effects were discussed in section 4.3.5.

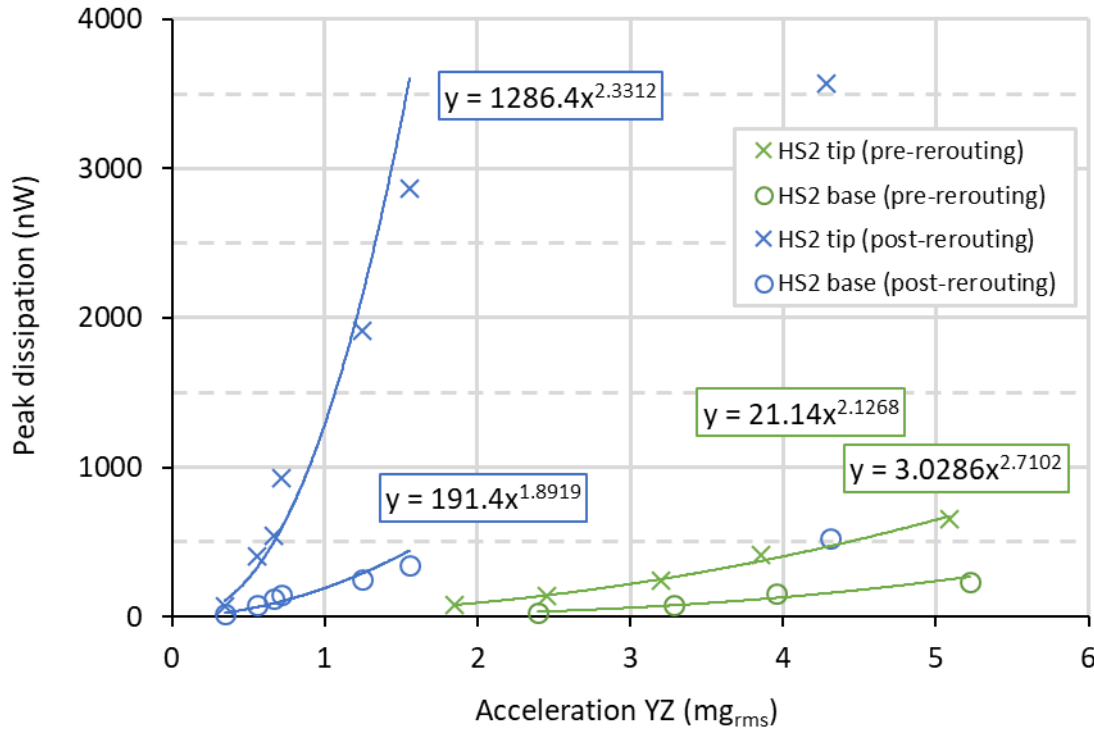


Figure 4.44 – Peak dissipation versus acceleration, HS2 base and tip, before and after wire rerouting (Vespel configuration)

It is also important to note that each data point corresponds to a peak dissipation and acceleration level at the same excitation frequency. Between each other though, the different points are not necessarily at the same excitation frequency. Even for a given test setup, we observe small differences in the peak dissipation frequency. For example, the dissipation peaks of the base of the heat switch after wire rerouting (see Figure 4.42) lie between 221.28 Hz and 221.32 Hz. This variation could be due to lags of the PID control used to regulate the temperatures, the physical time constants of our test setup or even variations in mechanical transmission of the cryostat. Here the acceleration level does not vary much over this 40 mHz range, which is different for configurations showing very high Q factors seen previously.

Taking this analysis further, we can now study the correlation with the model over the full excitation frequency band rather than only at the frequency of peak dissipation.

4.4.3.2 Full frequency band

Focusing on the dissipation levels observed at the heat switch tip, we can normalise them to the square of the measured acceleration level a^2 over the full excitation frequency band. If there is indeed perfect proportionality between microvibration dissipation and the square of the injected acceleration level, all curves should overlap irrespective of the excitation level. We can also vary the exponent value n in a^n to deduce an optimal value where they appear to overlap most.

We begin with the dissipation levels *before* wire rerouting (see Figure 4.41) and normalise them for 4 different exponent values: 1.5, 2.0, 2.5 and 3.0. Each is shown in Figure 4.45, in units of nW/mg^n . We neglect the 2 lowest excitation levels (30 mV_{rms} and 50 mV_{rms}) that were very noisy after normalisation.

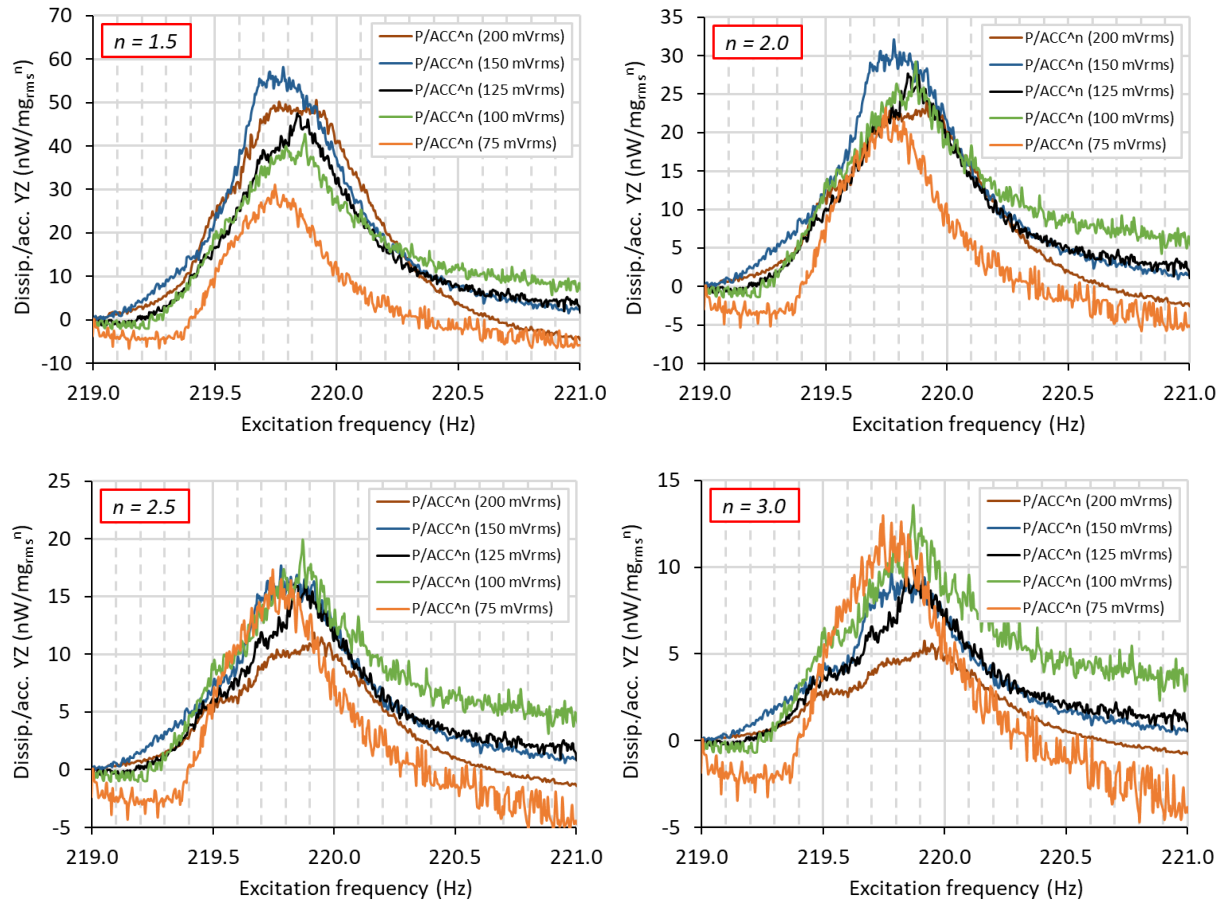


Figure 4.45 – Dissipation at HS2 tip *before* wire rerouting, normalised to injected acceleration level to the power of n for: $n = 1.5$ (top left), $n = 2.0$ (top right), $n = 2.5$ (bottom left) and $n = 3.0$ (bottom right)

For $n = 1.5$, the dissipation curves are almost perfectly in the order of excitation: higher excitation leads to higher $\text{nW/mg}^{1.5}$ values. The opposite is true for $n = 3.0$, where we note an almost perfect inversion of the order of excitation: higher excitation leads to lower $\text{nW/mg}^{3.0}$ values. This suggests that the optimal overlap (and so descriptor of the relationship between dissipation and injected acceleration level) is for $1.5 < n < 3.0$. In the 2 remaining plots ($n = 2.0$ and $n = 2.5$), we see good overlap. The optimal n value between 2.5 and 2.0 is to be compared to the corresponding power law fit of Figure 4.44 that has an exponent value of 2.3. We conclude that both methods are in agreement with each other and more importantly, in agreement with the dissipation model that predicts an exponent value of 2.

For comparison, we can also carry out the same method of normalisation of the dissipation curves for the tip of the heat switch *after* wire rerouting (see Figure 4.42). We now choose the same exponent values of 1.5, 2.0, 2.5 and 3.0. Each is shown in Figure 4.46, also in units of nW/mg^n . We neglect the highest excitation level (50 mV_{rms}) as it was very noisy after normalisation.

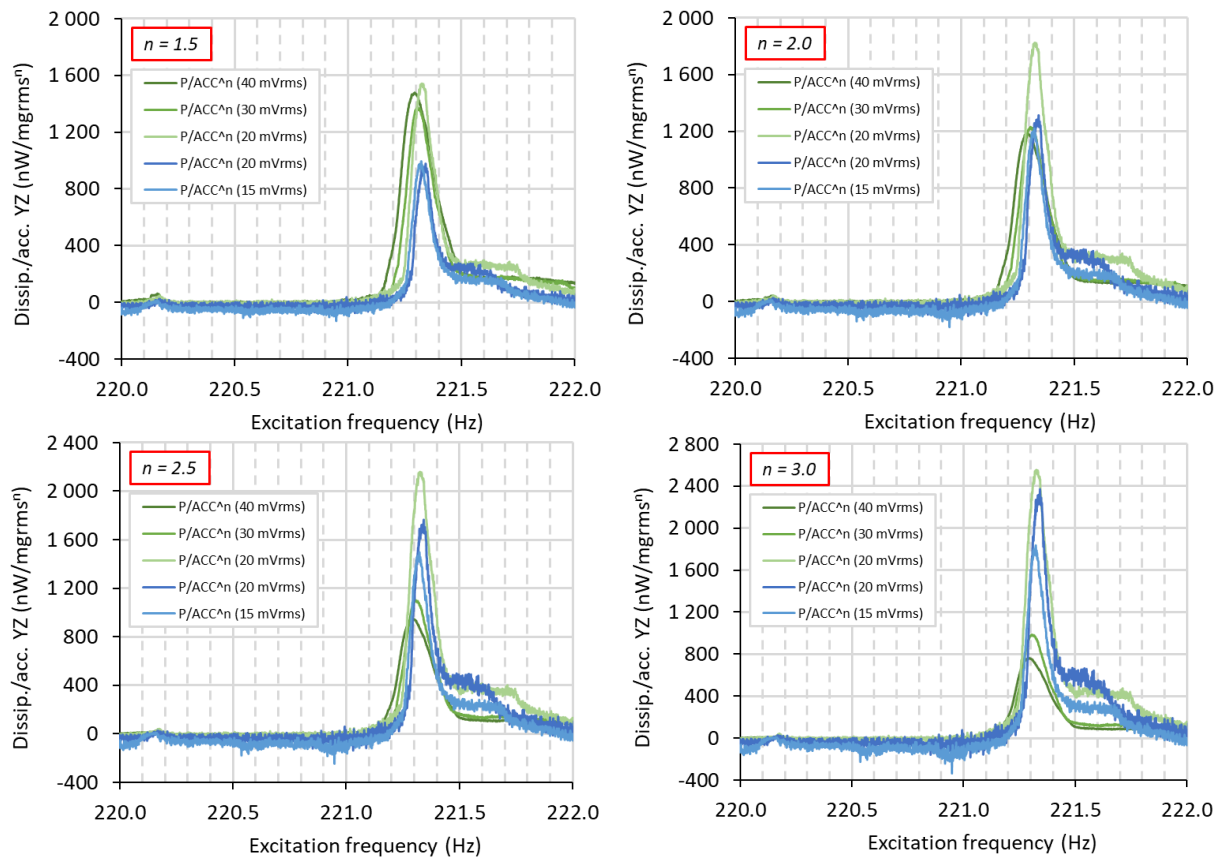


Figure 4.46 – Dissipation at HS2 tip after wire rerouting, normalised to injected acceleration level to the power of n for: $n = 1.5$ (top left), $n = 2.0$ (top right), $n = 2.5$ (bottom left) and $n = 3.0$ (bottom right)

Again, the dissipation curves are roughly in order of excitation for $n = 1.5$ and in the opposite order for $n = 3.0$. The excitation at 20 mV_{rms} appears to be an outlier in each of the 4 cases. Omitting this excitation level, the dissipation curves overlap well for $n = 2.0$, at around 1200 nW/mg². This optimal exponent value is close to the corresponding power law fit of Figure 4.44 that has an exponent value of 2.3. We can again conclude that both methods of deducing the exponent value are in agreement with the model prediction.

We have now explored the validity of the square power law relationship between the injected acceleration level and microvibration dissipation level, as predicted by the dissipation model. Given the measured system parameters (mass, Q factor, resonance frequency and injected acceleration level), it is now also worth studying the accuracy of the model in predicting the dissipation curves.

4.4.3.3 Predicted dissipation

As done in section 4.1.7 for the preliminary measurements, we can use measured system parameters as inputs to the model, to compare predicted versus measured dissipation. Using the scaling factor α , we fit our model to our measurements. We recall that α is adjusted to fit the peak dissipation levels.

Here we attempt to predict dissipation for a curve from each of the 3 cases:

- Vespel configuration before wire rerouting (see Figure 4.41)
- Vespel configuration after wire rerouting (see Figure 4.42)
- Steel washer configuration after wire rerouting (see Figure 4.43)

For the Vespel configuration, almost all dissipation appeared at the heat switch tip. For those 2 cases we therefore focused on the dissipation there. For the steel washer configuration however, we considered dissipation at both the base and tip, summing them. We must bear in mind though that, as we have seen, we likely underestimate the dissipation level at each of these locations as they probably occur before the base (in the steel washers) and after the tip (in the instrumentation wiring).

As input to the dissipation model, we use the apparent Q factors and resonance frequencies deduced from the dissipation curves, an estimation of the total mass at the tip of the heat switch and the acceleration levels measured at the cold plate (see section 3.5.1).

Beginning with the measurements *before* wire rerouting and in the Vespel configuration, Figure 4.47 shows the highest measured acceleration level (brown dotted line) and corresponding microvibration dissipation level (brown solid line) observed at the tip of the heat switch.

As per Figure 4.41, the measured Q factor for this dissipation curve is 406. We estimate the tip mass at 16.8 g. With an imposed reduction factor α (introduced in section 4.1.7) we obtain a predicted dissipation curve (red solid line). Here we must use a large scaling factor ($\alpha = 10.5$) to fit the model to the data.

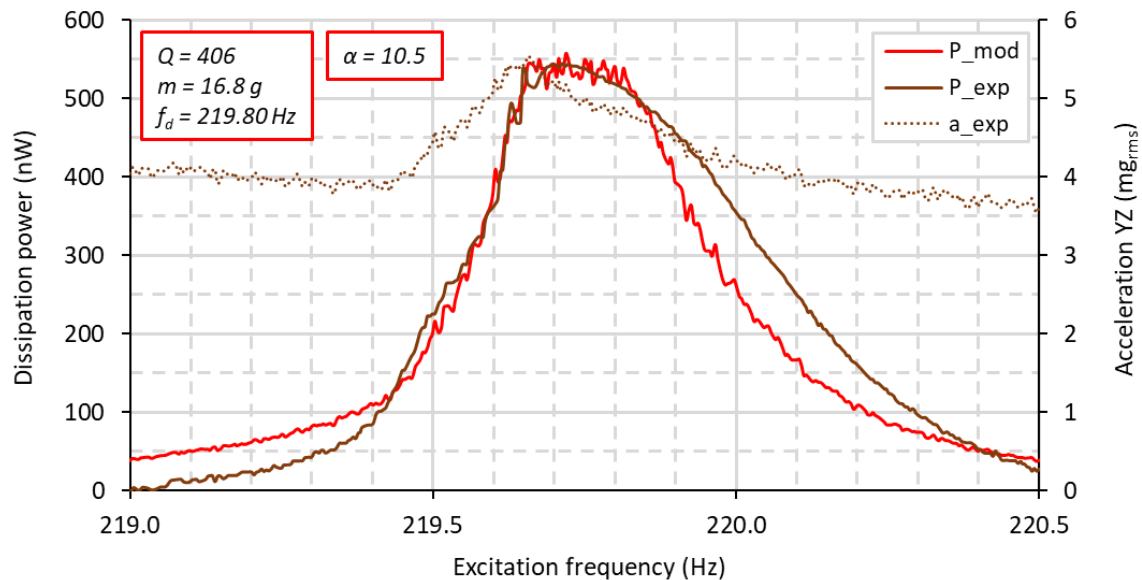


Figure 4.47 – Model fit (with large scaling factor) to observed dissipation of HS2 tip, before wire rerouting in Vespel configuration (shaker drive voltage of 200 mV_{rms})

We repeat the same procedure with the highest excitation level and dissipation curve *after* wire rerouting in the Vespel configuration. In this case, Figure 4.48 shows the highest measured acceleration level (black dotted line) and corresponding microvibration dissipation level (black solid line) observed at the tip of the heat switch. As per Figure 4.42, the measured Q factor for this dissipation curve is 1005. We keep the tip mass at 16.8 g. We then obtain a predicted dissipation curve (red solid line). Here, the model predicts the observed dissipation curves well for roughly $\alpha = 2.9$.

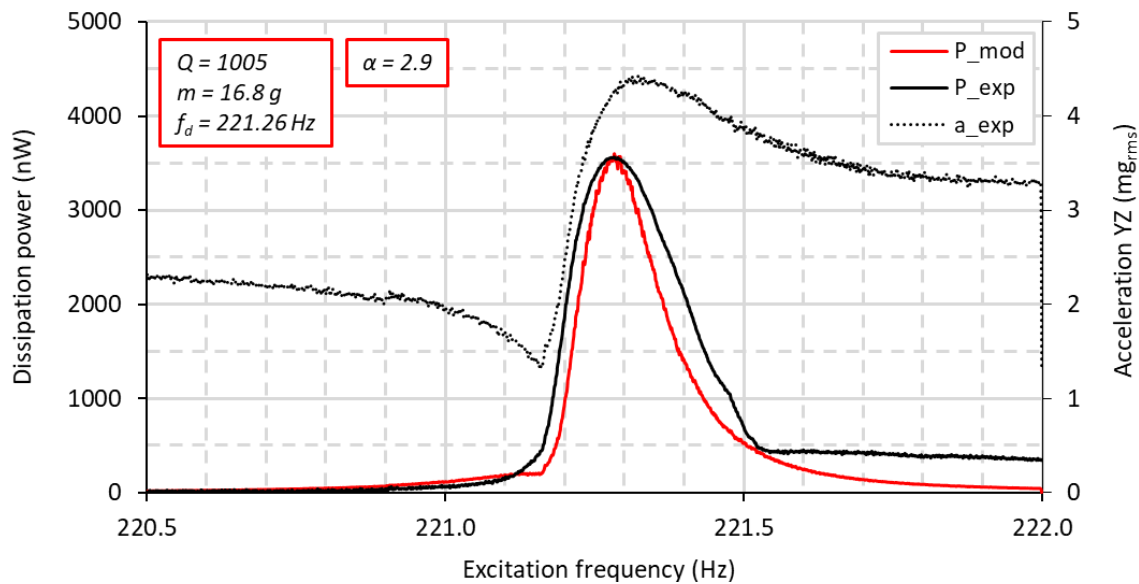


Figure 4.48 – Model fit (with scaling factor) to observed dissipation of HS2 tip, after wire rerouting in Vespel configuration (shaker drive voltage of 50 mV_{rms})

Lastly, we compare the model to dissipation observed with the steel washer configuration. For the 2 cases we have just seen, only dissipation at the tip of the heat switch was considered for correlation with the model, as most of the total dissipation appeared there. In the steel washer configuration however, most appears at the base. The nature of this dissipation affects how we choose how to consider it for the model correlation. Figure 4.49 shows two cases of dissipation that would cause a thermal effect at the base.

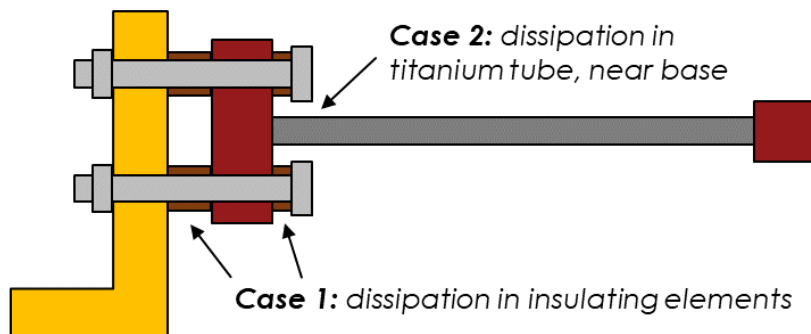


Figure 4.49 – Possible causes of dissipation seen at the base of the heat switch (insulated base configuration)

A first possibility is that dissipation is occurring in the insulating elements rather than in the heat switch itself, as we have previously discussed. If this is due to relative motion between the bracket and base, then the behaviour of our system is closer to a 2-DOF mass-spring than the 1-DOF one modelled. In that case, for correlation with the model, we would only consider the dissipation level that appears at the tip.

A second explanation is that dissipation is occurring in the titanium tube (where mechanical stress is greatest) as originally hypothesised, or even in the brazing between it and the base of the heat switch. In this case, the relative motion between the bracket and base may be zero, such that our system more closely behaves like a 1-DOF mass-spring. For model correlation, we can then consider the *sum* of the dissipation levels at the base and tip.

For the sake of the correlation with the model, we will assume the second case and so consider the sum of the observed dissipation levels. This may not be representative of the nature of the dissipation due to microvibrations observed at the base, though allows us to explore the limits of our dissipation model.

From the steel washer dissipation tests, we take the case of a drive voltage of 80 mV_{rms}. We do not choose the highest excitation level case (60 mV_{rms}) as the corresponding dissipation curve at the tip is distorted (see Figure 4.43). We observe higher dissipation for a lower shaker drive voltage of 60 mV_{rms} as the acceleration level is also higher at this excitation level. This could be explained by the time span between the tests, during which the cryogen reservoirs may have emptied slightly, affecting the mechanical transmission of the cryostat (these aspects were discussed in section 3.1).

Figure 4.50 shows the measured acceleration level (green dotted line) and corresponding total microvibration dissipation level (green solid line) measured in the heat switch, for the 80 mV_{rms} case. The dissipation curve is shifted by 11 mHz (to the left) to align its peak with the acceleration peak, as we believe this apparent shift is a measurement error due to the chosen sweep speed (1000 s/Hz) rather than a phenomenological one. This was previously discussed in section 3.4.2.3). This adjustment was not needed for the previous model fits as the Q factors were sufficiently low and so the bandwidths higher.

As per Figure 4.43, the Q factor for the observed dissipation curve is 4070. The increase in the Q factor increases the distortion of the dissipation curve and so the error in the calculation of the Q factor beyond the previously acceptable level of 20%. We know that the Q factor has increased considerably, as the mechanical feedback of the heat switch resonance clearly shows in Figure 3.17, though we now cannot deduce its value precisely. For the model input we adjust its value to 2000 and keep a tip mass of 16.8 g.

With these system parameters we obtain predicted dissipation (red solid line). Here, the model predicts dissipation for $\alpha = 9.0$. Thus our model still overestimates the measured dissipation level.

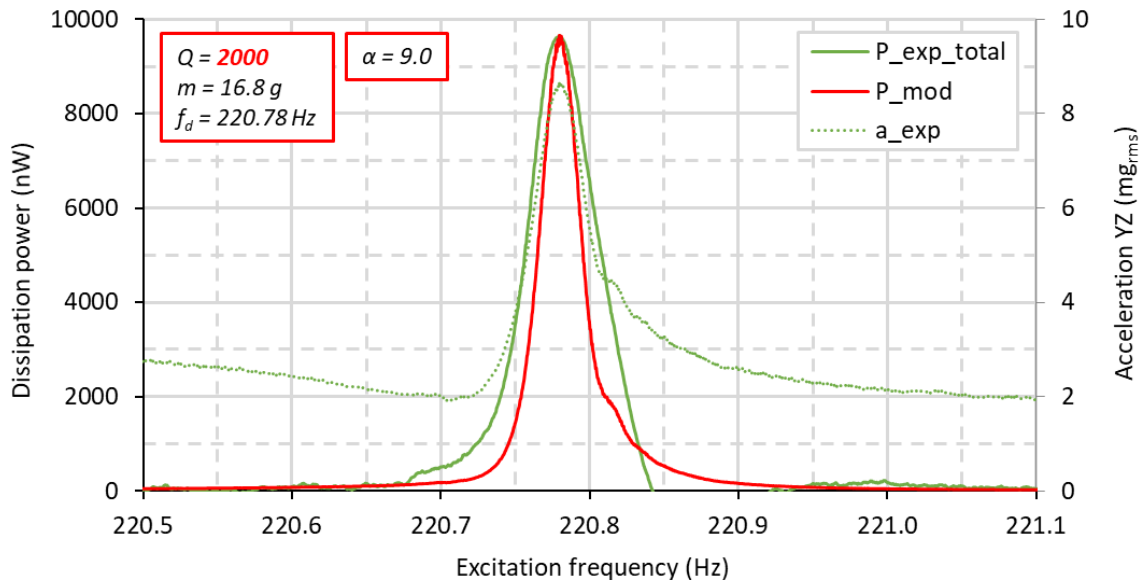


Figure 4.50 – Model fit (with scaling factor) to observed dissipation of HS2 base, in steel washer configuration (shaker drive voltage of 80 mV_{rms})

Having now presented our attempts to fit the model to measured dissipation curves using measured input parameters, it is worth discussing the key takeaways of this subsection 4.4.3 on model correlation.

4.4.3.4 Discussion

We will now discuss the various ways in which we studied the correlation between our experimental data and the dissipation model, and summarise the key takeaways.

Section 4.4.3.1 first presented power law fits of peak dissipation versus injected acceleration level. We studied these fits for dissipation at the base and tip of the heat switch, and before and after wire rerouting. For the 4 cases, we found exponent values between 1.9 and 2.7. The exponent value of 2.7 corresponds to the case of lowest observed dissipation (at the base, before wire rerouting). These exponent values are compatible with the model, which predicts a square power law such that $P \propto a^2$.

Though we were not able to confirm the possibility during our tests and analyses, as our dissipation model only incorporates viscous damping (i.e. proportional to velocity), it could be representative of the dominant dissipation mechanism. The various forms of damping that may exist were discussed in section 0.

We should also reflect on the precision of these fits however. We assume that all other system parameters are constant. We have now seen though, that the Q factor generally decreases with increasing mechanical excitation. As per equation (1.45) of chapter 1, this directly influences the dissipation level. Furthermore, threshold effects were not accounted for (the power law fits passed through the origin). If present, non-zero acceleration levels that cause no dissipation would steepen alter a power law fit.

Next in section 4.4.3.2, we studied the square power law over the full excitation frequency band. We focused on dissipation at the tip of the heat switch and in the Vespel configuration only. Normalising the dissipation curves to $a^{1.5}$, $a^{2.0}$, $a^{2.5}$ and $a^{3.0}$, we deduced optimal exponent values between 2.0 and 2.5. This is also compatible with the square power law relationship predicted by the dissipation model. We must note nevertheless, that the same imprecisions apply here as for the power law fits above.

Furthermore, comparing the normalisation of the dissipation curves to $a^{2.0}$, both before and after wire rerouting, we once observe a considerable increase: from roughly 25 nW/mg² (before rerouting) to 1200 nW/mg² (after rerouting). This reinforces one of the conclusions of section 4.4.1, which is that wire routing strongly impacts microvibration dissipation levels.

Lastly, in section 4.4.3.3, measured parameters were used as inputs to the model to predict microvibration dissipation. A reduction factor α was used to fit the model to the measured dissipation level. We selected a dissipation curve for each of the 3 cases: in the Vespel configuration before and after wire rerouting, and in the steel washer configuration. Prior to wire rerouting the α value was 10.5, whereas after wire rerouting it was 2.9. In the steel washer configuration, it was 9.0. Thus in all 3 cases (in addition to section 4.1.7 where $\alpha = 5.9$), the model strongly overestimates the measured microvibration dissipation level.

An overestimation of the model predictions may be caused by an overestimation of the system parameters, in particular the Q factor. As detailed in chapter 3, the apparent Q factor deduced from a dissipation curve is not precise as the latter is distorted due to an injected acceleration profile that is not flat. This is not problematic as long as the apparent Q factor is not too high and so the mechanical feedback from the heat switch to the cold plate is not either. For the final case studied however, namely the dissipation curves of section 4.4.2, the apparent Q factors exceeded 5000. The estimation of the Q factor (and so the fitting of the dissipation curve to the model) are then imprecise. This highlights

limitations of the measurement environment (unwanted mechanical feedback) but also of the model, as simply correcting the Q factor as input parameter still does not enable proper fitting of the model.

However, what appears to be an overestimation of the model may in fact be an underestimation of the measured dissipation. As amply discussed throughout this chapter, if dissipation occurs elsewhere, say in the wiring of the instrumentation due to its oscillation and deformation, only part of the heat load will appear at the heat switch tip. If indeed the case, the true α values would then be lower.

4.4.4 Accelerometer at tip

As a final test setup to study microvibration dissipation, we evolved the steel washer configuration by adding an accelerometer to the heat switch tip. This is shown in Figure 4.51.

The main motivation for doing this was to have access to another method of measuring the Q factor at low temperatures, namely through the mechanical amplification at the tip. This can then be compared to deducing the Q factor from the dissipation curves using the bandwidth definition (see section 1.3.7.1). Until now, we recall that the latter was the only method of deducing the Q factor at low temperatures.

Measuring mechanical amplification using an accelerometer at the tip is possible at ambient temperature too. We can therefore observe the impact of cooling on the Q factor more reliably, using a single method. This was previously only possible using different methods to measure it at ambient and low temperature.

Another aspect worth studying is changes (if any) in the dissipative behaviour of the test setup, given the changes we have made to its mechanical behaviour (e.g. through added mass or altered damping).

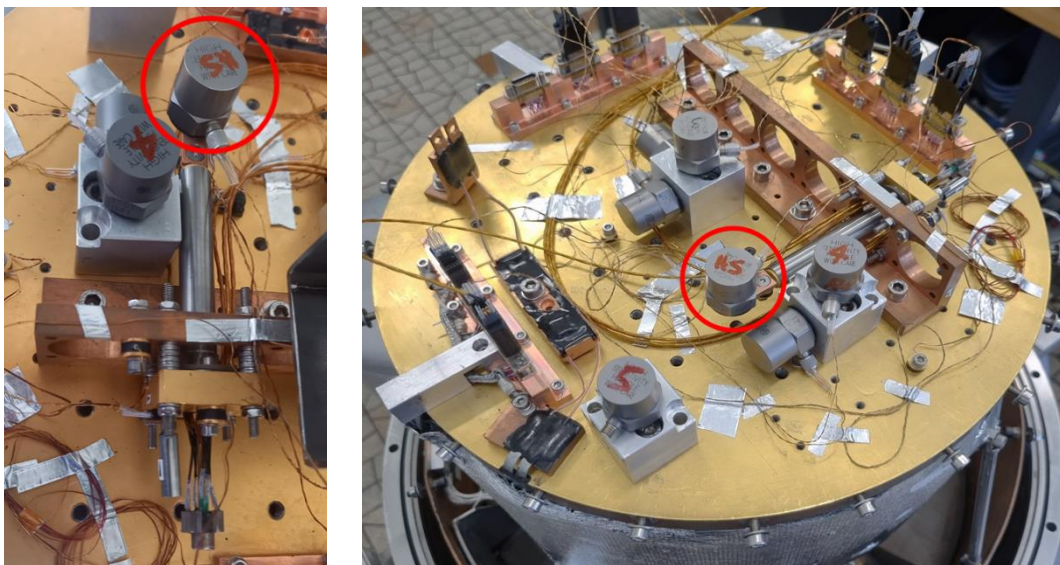


Figure 4.51 – Accelerometer mounted to HS2 in steel washer configuration

4.4.4.1 Impact of cold on Q factor

We first explore the impact of cooling on the Q factor measured through the mechanical amplification of the heat switch tip. We perform acceleration measurements at 300 K and 1.2 K.

The mechanical amplification is estimated by taking the ratio of the acceleration level at the tip to its level at the cold plate. We measure this gain for sweeps from 20 Hz to 200 Hz (the added mass is expected to reduce the resonance to below 200 Hz) and a shaker drive voltage of 200 mV_{rms} (the 10X voltage divider was used for all tests here of section 4.4.4). The gains are shown in Figure 4.52.

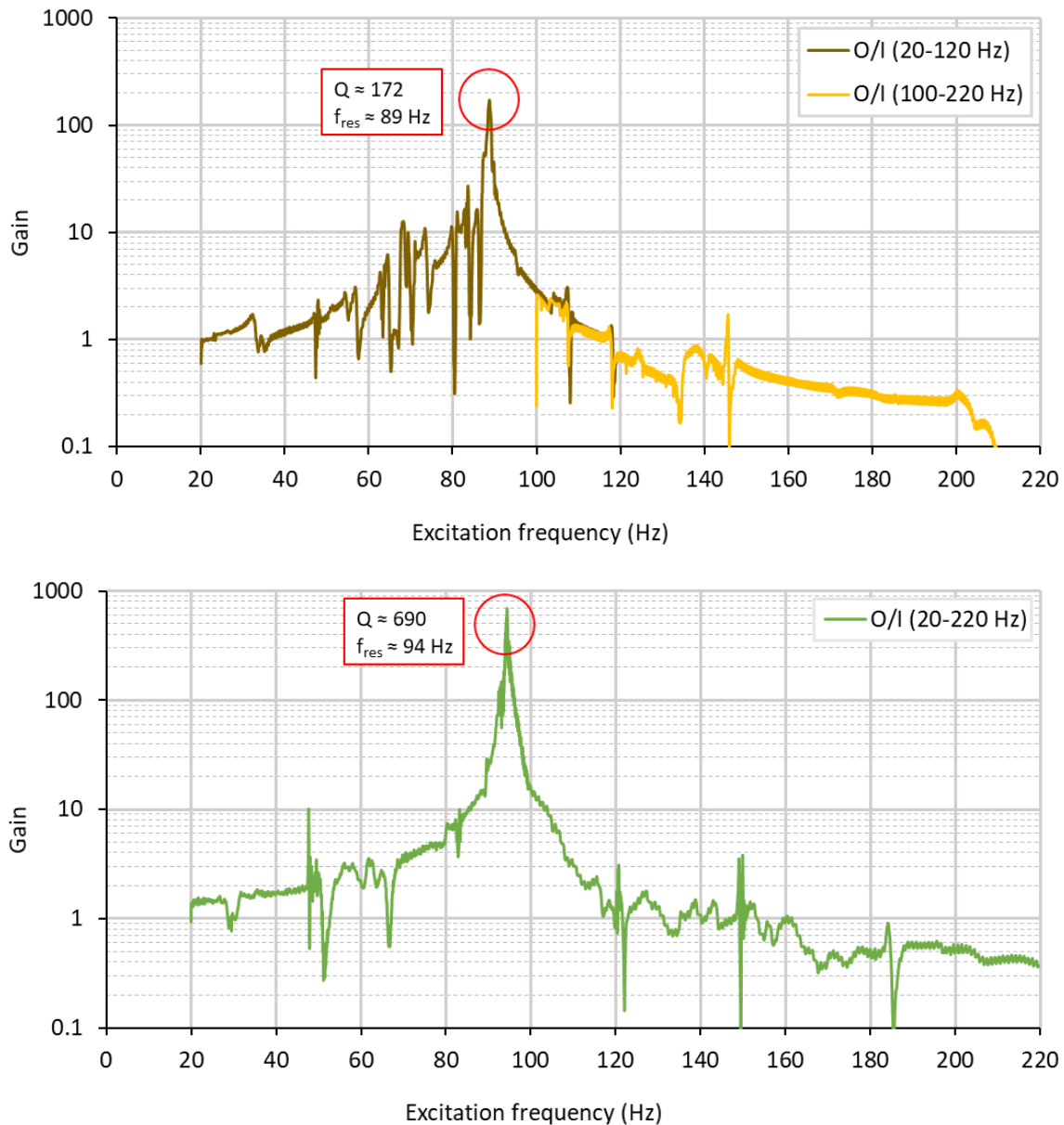


Figure 4.52 – Mechanical amplification of HS2 when subjected to vibrations at ambient temperature (top) and 1.2 K (bottom), for a shaker drive voltage of 200 mV_{rms} and no temperature regulation (steel washer configuration with accelerometer at tip)

At 300 K, the Q factor was 172. At 1.2 K, it rose by a factor 4 to 690. This is in agreement with the evolution of the Q factor seen in other literature, discussed in section 1.2.2 The Q factor thus appears to be much lower (even at 1.2 K) than for the steel washer configuration with no accelerometer at the tip (over 2000).

As expected the resonance frequency decreases a lot. From 300 K to 1.2 K it increases by 5 Hz, indicating higher stiffness as is expected when the titanium tube is cooled, owing to a higher Young modulus.

At the resonance, the drive voltage of 200 mV_{rms} corresponds to a cold plate acceleration of ~ 0.4 mg_{rms}. Over the full 20-220 Hz excitation frequency range however, this acceleration level spans ~ 2 orders of magnitude, as was also observed in section 3.1.1.

We can now compare the cold Q factor deduced mechanically to the microvibration dissipation response curves obtained using the same setup, with an accelerometer at the tip.

4.4.4.2 Reduced Q factor

The dissipation measurements at 1.2 K are shown in Figure 4.53, the accelerometer still mounted to the tip. Again, we observe more dissipation at the base of the heat switch, though dissipation levels have now considerably reduced and the Q factor has dropped from at least 2000 (steel washer configuration with no accelerometer at the tip) to ~ 600 here.

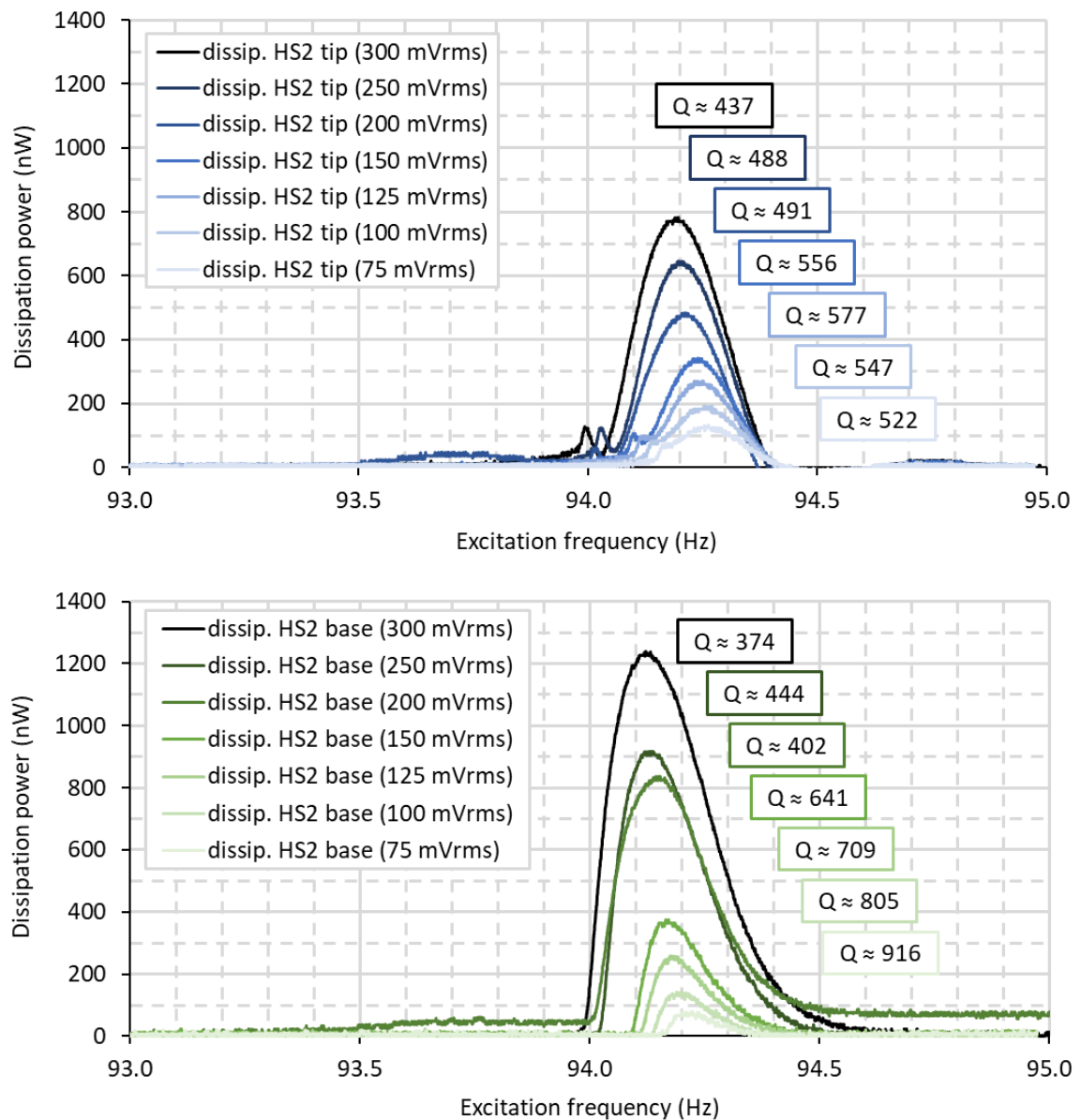


Figure 4.53 – Microvibration dissipation levels at tip (top) and base (bottom) of HS2 with accelerometer at tip at 1.2 K

4.4.4.3 Analysis

Regarding the dissipation levels, these have now reduced to $\sim 1.2 \mu\text{W}$ at the base and 800 nW at the tip (thus still more at the base) from levels exceeding $11 \mu\text{W}$ when no accelerometer was mounted to the tip. This is owed to the considerable decrease in the Q factor, from values exceeding 2000 back down to ~ 600 .

This reduction in the Q factor is likely explained by the addition of the new wiring of the accelerometer at the tip. Indeed this wiring may be a source of damping, reducing the purity of the resonance (a lower Q factor) of our system. Another reason for this decrease could be due to the decrease by a factor 2 in the resonance frequency. As previously discussed, displacement amplitude is inversely proportional to the square of the excitation frequency, for a given acceleration level. If certain dissipative phenomena only appear at higher displacement amplitudes, which have increased here by a factor 4, we can therefore expect lower Q factors for higher displacement values that occur due to lower resonance frequencies.

Despite the mechanical and thermal measurements not being carried out synchronously (the added heat load of the accelerometer at the insulated tip was too high for reasonable temperature regulation), the deduced Q factors are in agreement with each other: ~ 600 deduced from the dissipation curves (Figure 4.53) and ~ 690 deduced mechanically from the acceleration measurements (Figure 4.52). This suggests that our method of deducing the Q factor from the dissipation curves is valid (as long as it is not too high).

4.5 Conclusion

This chapter has presented the microvibration dissipation measurements carried out on gas-gap heat switches and the analyses thereof. The focus was on dissipation at low temperatures (4.2 K and 1.2 K), complemented with noteworthy observations at 300 K. We conclude this chapter with the key takeaways.

We first presented preliminary measurements made using the simplified test setup, where we directly mounted the base of the heat switch to the bracket. Through an elevation of the temperature at the tip, we observed the dissipation due to microvibrations. This dissipation occurred at distinct fixed excitation frequencies, reaching levels of at least $2 \mu\text{W}$ for injected acceleration levels of $4 \text{ mg}_{\text{rms}}$. Dissipation appeared over a very narrow frequency range of $\sim 0.6 \text{ Hz}$ (i.e. a very low bandwidth), due to Q factors exceeding 1000 for these tests.

We presented the limitations of this configuration (where the base of the heat switch was not insulated) through analytical examples. We detailed that what appeared to be measured dissipation at the tip could be only a part (i.e. an underestimation) of the total dissipation occurring in the heat switch. We justified the evolution of the test setup and presented several insulated base configurations. These findings enabled us to deduce the total microvibration dissipation occurring in the heat switch and partially informed us on its location. This in turn allowed us to present the advanced measurements of this thesis.

The heat switch base now insulated, we expected most of the dissipation to appear there rather than at the tip, though most still appeared at the tip. We theorised that bending stress in the titanium tube was not the primary dissipative phenomenon. We then hypothesised that the oscillations of the instrumentation wiring attached to the tip of the heat switch was causing high levels of microvibration dissipation.

The test setup was adapted to study this. When we removed instrumentation from the tip of the heat switch, the Q factor at 300 K increased by a factor 2.5 (~ 140 to ~ 350). This suggested the instrumentation and its associated wiring have a big impact on the Q factor and so on the dissipative behaviour of our test setup. To study this at low temperatures however, the tip had to be instrumented. Rerouting the wiring carefully, we achieved a Q factor at 300 K of ~ 300 , close to the Q factor without instrumentation wiring.

We also found an impact of this wire rerouting on the Q factor and dissipation. At low temperatures, the calculated Q factor rose from ~ 400 to ~ 2400 . This caused dissipation bandwidth to fall sharply and peak dissipation to rise from ~ 500 nW for $5 \text{ mg}_{\text{rms}}$ of mechanical excitation to ~ 3.5 μW for $4 \text{ mg}_{\text{rms}}$.

The discrepancies in Q factors and dissipation levels between all our tests highlights the susceptibility of our dissipation measurements not only to our wire routing choice (an aspect that we did not consider during the preliminary dissipation measurements) but also to our test configuration.

To study the impact of the test configuration, we also made dissipation measurements with the steel washer insulation, replacing the Vespel elements. Contrary to all previous tests, we observed more dissipation at the base of the heat switch than at the tip. This suggested that the insulating elements themselves may be dissipating microvibrations, for example by rubbing against each other. The apparent Q factor also increased considerably, exceeding values of 4000. We explained why this was likely an overestimation of the true value and proposed to use a value of 2000 for correlation with the model.

Several of the presented dissipation measurements were then correlated with the dissipation model that was presented in chapter 1. The relationship between dissipation and injected acceleration was explored at peak dissipation (i.e. at a single excitation frequency) but also over the full excitation bandwidth. Fitting power laws, we found close agreement with the square power law predicted by the model, between dissipation and injected acceleration (that is to say that $P \propto a^2$). We also compared the dissipation model to dissipation curves, using as model inputs measured system parameters. The model always appeared to overestimate the measured dissipation level (a reduction factor α ranging between 2.9 and 10.5). However, we proposed that what appears to be an overestimation of the model could in fact be an underestimation of the measured dissipation level. As previously discussed, if dissipation in fact occurs elsewhere than where we regulate the temperatures, its level is necessarily underestimated. However, since we cannot verify this, it remains necessary to find a conclusive demonstration for this explanation.

As a final series of tests, we mounted an accelerometer to the tip of the heat switch in the steel washer configuration. We could now directly compare the Q factor deduced thermally with a way of deducing it mechanically through the amplification of the oscillations. At low temperatures, the 2 methods were in agreement with each other, with observed Q factor values of ~ 600 . We could also compare the evolution of the Q factor from hot to cold: we found a roughly fourfold increase from ~ 172 to ~ 690 . The Q factor has fallen significantly, from apparent values exceeding 5000 before mounting the accelerometer. This agrees with our conclusion that adding wiring (now of the accelerometer at the tip) reduces the Q factor.

To summarise, when the measurements are ideal and especially at low temperatures where the Q factor is higher, the heat switch itself shows little dissipation with Q factors exceeding 2000. Wiring attached to its tip reduces the Q factor to below 600. Objects of negligible mass compared to the heat switch and that introduce damping (e.g. wiring), can therefore contribute to dissipation observed at the heat switch resonance that lies beyond their own resonances. This may occur due to the forced oscillation of one end of a dissipative object such as a wire that is attached to the heat switch, the other end remaining fixed.

Conclusion and perspectives

The main goal of this thesis was to further the understanding of the phenomena causing microvibration dissipation at low temperatures. My work aimed to shed light on the performance degradation of sub-Kelvin thermal detectors due to this dissipation, in future space missions studying the science of the universe (e.g. Athena), with lessons learnt from past ones (e.g. Planck and Astro-H).

My work combined mechanical and thermal measurements, with the added difficulty of operating in cryogenic conditions. Despite this challenging test environment, we successfully designed a test bench dedicated to microvibration dissipation thanks to carefully chosen mechanical and thermal equipment, and proposed bespoke measurement methods.

Following the tests on a complex cryogenic system (an ADR and sorption based sub-Kelvin cooler), we found that we could benefit from studying a simpler system, namely an elementary cryogenic device (a gas-gap heat switch), to gain a more fundamental understanding of microvibration dissipation.

Our custom sensing and actuation chain was able to induce sub-mg microvibrations and measure sub-microwatt levels of microvibration dissipation, in cryogenic vacuum conditions at 1.2 K and 4.2 K. Dissipation was measured when exciting close to the mechanical resonance frequency of our test object. As an order of magnitude, for 1 mg_{rms} of mechanical excitation we observed dissipation as high as 1 μ W.

Exploiting our test environment to its fullest we were also able to identify factors influencing its mechanical behaviour. We found that the cryostat is not mechanically ideal, observing mostly unpredictable changes. We identified large impacts from the cryogen mass and working temperature, as well as mechanical feedback from our test object. These aspects add complexity to the knowledge and control of the environment and so to the proper characterisation of observed microvibration dissipation.

Despite this imperfect environment, we were able to attribute large variations in the Q factor (both at ambient and at cryogenic temperatures) to changes in the routing of instrumentation wiring, as well as to the insulating elements used to thermally decouple the heat switch from the cold plate. When bare, the heat switch itself showed a high Q factor. Introducing surrounding components decreased the Q factor, indicating that compared to the heat switch they introduce significant damping (they are more dissipative). Indeed, rather than finding dissipation in the heat switch itself, it was mostly in surrounding components.

Using only thermometers and heaters, we also managed to measure the mechanical resonance of our test object and deduce Q factors exceeding 2000. Measuring these may not have been possible using an accelerometer, as introducing it was found to reduce the Q factor and shift the resonance (due to added mass). We found that for higher Q factors, mechanical feedback introduces imprecision in the knowledge of the input acceleration, reducing the accuracy with which we calculate the Q factor.

We used a dissipation model (based on viscous damping in a mass-spring system) as a useful tool to analyse our results. The model predicts that reduced damping (a higher Q factor) leads to higher peak dissipation over a reduced frequency range (a lower bandwidth) centred at the resonance. This is also observed through our experiments. Counterintuitively, this amounts to saying that the removal of a dissipative element in fact *increases* the maximum of the dissipation response.

From our dissipation model we derived an expression of this peak dissipation level at resonance. It is a good tool to estimate it and complements our experimental values:

$$P_{max} = \frac{mQ}{2\omega_0} a^2$$

The proportionality our dissipation model predicts between the square of the injected acceleration level and dissipation (not only at resonance) is in agreement with most of our experimental observations. This suggests that viscous damping is representative of the dominant dissipation mechanism.

We also used a thermal model to show that the location of dissipation does not always correspond to the location where the thermal changes are observed. The thermal impact measured at say the tip of the heat switch, of dissipation occurring in the instrumentation wiring, is only an *image* of it. In such cases, we showed that we always *underestimate* the true microvibration dissipation level. This could explain why our dissipation model systematically predicted higher dissipation levels than we appeared to measure.

Now too, at the end of our studies with the gas-gap heat switch, we realise that we would gain valuable insights studying simpler systems still. We could design a simpler geometry (say a pure cantilever beam) or even study microvibration dissipation in single wires. As our objects under study reduce in size, special attention would have to be paid to less invasive instrumentation too.

These include non-contact methods of measuring mechanical amplification (as well as displacement and acceleration) such as capacitive or fibre optic sensors. The latter is expensive, requires significant modifications to our cryostat (partly the reason for not having explored this option) and may even introduce an additional heat load onto our test object. It would give us “eyes in all directions” however, allowing us to explore all the degrees-of-freedom of our test object. Non-contact temperature measurement in cryogenic conditions could also be explored.

Other common building blocks of cryogenic systems could be studied too, such as thermal straps, insulating structural supports (typically made of CFRP) or insulating suspension systems. This would give us a more complete view of microvibration dissipation in cryogenic systems. We have in fact built a new test object – a Kevlar string suspension system – but have not tested it for microvibration dissipation yet. It is a mockup of an ADR cooling stage and was presented earlier in the dissertation. The methods presented here will be helpful in conducting the measurements and analyses for this object.

To improve the stability of injected microvibration levels, our test object could be mounted rigidly, on a high-mass mechanical test bench. Cooling is then impractical however. A more viable solution would be to use a heavier and more rigid cryostat that is mechanically isolated from the ground. Inducing microvibrations much closer to the test object interface (the cold plate) than currently (on the exterior of the cryostat) would also limit the influence of the cryostat transmission (shown to vary by ~2 orders of magnitude). For this, we need a cold actuator (that functions at cryogenic temperatures). In this way, the injected microvibration level would be flatter across the excitation frequency range. Remaining variations of the acceleration level could be stabilised further by means of PID control.

We may also explore microvibration dissipation through a more fundamental study of material properties. It is worthwhile gaining a deeper understanding of the mechanical damping and dissipative behaviour of different materials when subjected to mechanical vibrations or deformation.

More broadly, our work also applies to complex cryogenic systems at spacecraft-level. For future low-temperature science missions such as Athena, microvibration dissipation threatens both the available cooling power (as low as $1\ \mu\text{W}$ at $50\ \text{mK}$) and the thermal stability (below $1\ \mu\text{K}$) of the coldest cooling stage. Ours findings are also relevant beyond space. For quantum computers for example, very low temperatures (as low as $10\ \text{mK}$) help maintain the quantum state of the qubits that they rely on. This so-called quantum coherence is also very sensitive to energy fluctuations such as microvibration dissipation.

Appendices

Appendix 1 – Stabilisation and cooling times

It can also be of interest to know the transient response of the heat switch, characterised by its thermal time constant. For the tip of the heat switch specifically, through the thermal time constant we can estimate the time needed to cool it from ambient temperature to say 4.2 K or to stabilise when subjected to a sudden heat load, say from the dissipation of microvibrations.

The time constant of the heat switch is the product of the thermal resistance R and thermal capacitance C , as discussed in the generalised thermal model of chapter 1. The resistance is mainly defined by the titanium tube, whereas the capacitance is mainly defined by the heat switch copper tip and the steel nuts and screws of the instrumentation there. The time constant is thus independent of the amplitude of the heat load that causes temperature elevations, though this is only true if we assume that these temperature elevations have negligible impact on the thermal properties of our system (i.e. R and C), which they in fact do. Bearing this in mind, we can still study stabilisation times if we assume temperature elevations are not large and so changes in thermal properties are negligible.

We choose to compare the time constants at 3 temperatures: 300 K, 4.2 K and 1.2 K. We also assume that the heat switch is in the OFF mode, thus conduction only take place in the titanium tube. Thermal properties of titanium, copper and steel that are used for our calculation at each temperature are listed in Table A. Together with the dimensions and masses of the components (presented in Table 2.1), we estimate the time constant at each temperature. The rightmost column of Table A lists them.

Table A – Theoretical thermal time constants for heat switch at different temperatures

	<i>Titanium conductivity</i>	<i>Copper heat capacity</i>	<i>Stainless steel heat capacity</i>	<i>Heat switch thermal time constant</i>
300 K	20 W/m/K	386 J/kg/K	477 J/kg/K	2h 21m 25s
4.2 K	2.85×10^{-1} W/m/K	1.00×10^{-1} J/kg/K	1.96×10^0 J/kg/K	8m 50s
1.2 K	5.71×10^{-2} W/m/K	1.47×10^{-2} J/kg/K	5.48×10^{-1} J/kg/K	10m 54s

We note different estimated time constants for the different temperatures, with a minimum closer to 4.2 K than 1.2 K. From the temperature elevations presented in section 4.1.2, we can deduce the time constants at low temperatures. From the generalised thermal model of chapter 1, we know that:

$$T(t) = T_2 + (T_1 - T_2) e^{-t/\tau}$$

Introducing the natural logarithm we can state:

$$\ln\left(\frac{T(t) - T_2}{T_1 - T_2}\right) = \frac{-t}{\tau}$$

Plotting the first member versus t , the slope is then equal to $-1/\tau$. Applied to the temperature elevation curves due to electrical heating of Figure 4.9 (that was at 1.2 K), we obtain the linear trends shown in Figure A. Here we only show the temperature decays following each heating phase as they were longer in duration than the temperature rises, which makes the estimation of the time constant more precise.

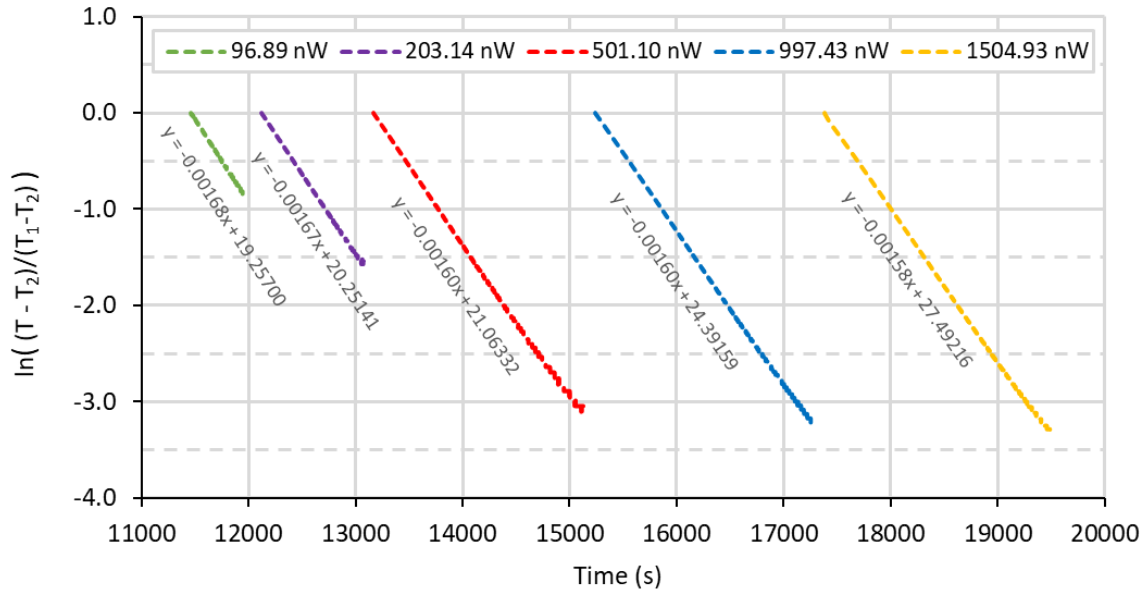


Figure A – Estimation of thermal time constant from temperature decay measurements at 1.2 K

The average slope is 0.0016, which corresponds to a time constant of 625 s (or 10m 25s). We perform the same procedure for the temperature decays at 4.2 K and obtain a time constant of 532 s (or 8m 52s). Hence the predicted time constants of Table A are in close agreement with those measured.

Regarding an estimation of the cooling time from say 300 K to 1.2 K, even for a simple thermal RC series circuit, the evolution of the thermal properties (i.e. R and C) with temperature must be accounted for. More importantly though, in the case of our heat switch, its vastly different conductances in the OFF mode and ON mode mean the estimation of its cooling time is complex. For our application, estimating the cooling time analytically has no real use (it is simply measured) and so will not be delved into.

Appendix 2 – Heat switches interaction

To study the mechanical interaction between the 2 heat switches when mounted to the same bracket, we measured the displacement of HS1 for different resonances of HS2. At 300 K the resonance of HS1 was 203 Hz and by adding or removing a dummy mass, the resonance of HS2 could be adjusted to roughly 210 Hz or 300 Hz. The impact on the displacement of HS1 is shown in Figure B.

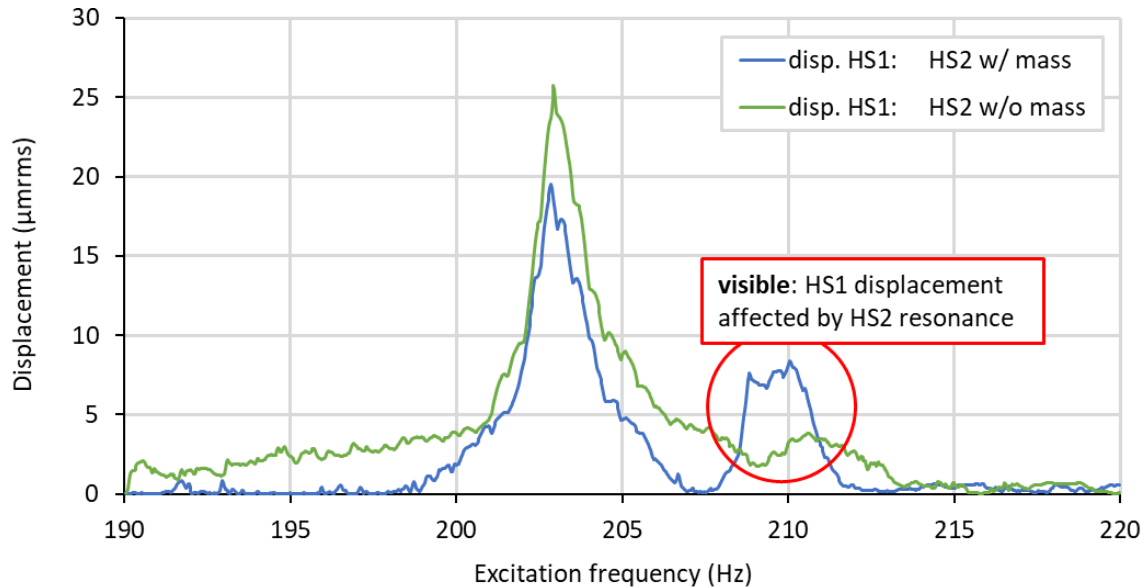


Figure B – Impact of resonance of HS2 on displacement of HS1

We observe here that the resonances are sufficiently narrow and the resonance frequencies sufficiently apart that the resonances do not overlap, despite only being about 7 Hz apart. In other words, the bandwidth (we take the FWHM value) of our test object must be less than the spectral distance between the 2 resonance frequencies. This implies that a higher Q factor is better, which contradicts the concluding remark of section 3.5 regarding the negative impact of mechanical feedback. To avoid the influence of one heat switch on the other and given the relatively low Q factors for parts of this thesis (of the order of 100 in some cases), we choose to mount only 1 heat switch at a time to the bracket when possible.

Appendix 3 – Tip load with dissipation profile

In chapter 4, different dissipation profiles were studied assuming zero dissipation at the tip of the heat switch and maximum dissipation at its base. These assumptions stem from the definition of the bending stress (which we believe could be a descriptor of microvibration dissipation) along a cantilever beam. Here we will show how even if only a small proportion of the total dissipation arrives at the heat switch tip, the power law describing the dissipation profile along the heat switch must have a very high order. We will study the case of the non-insulated heat switch base such that $T_0 = T_1$.

We impose the total dissipation P_{total} occurring in the heat switch to $10 \mu\text{W}$ and the cold plate temperature T_0 (and so tip temperature T_1) to 1.2 K . For a power law profile of order 2, with no heat load at the tip of the heat switch, the tip temperature T_2 is 1.875 K . We shall take this as the reference case. This dissipation profile and resulting temperature profile is shown Figure C.

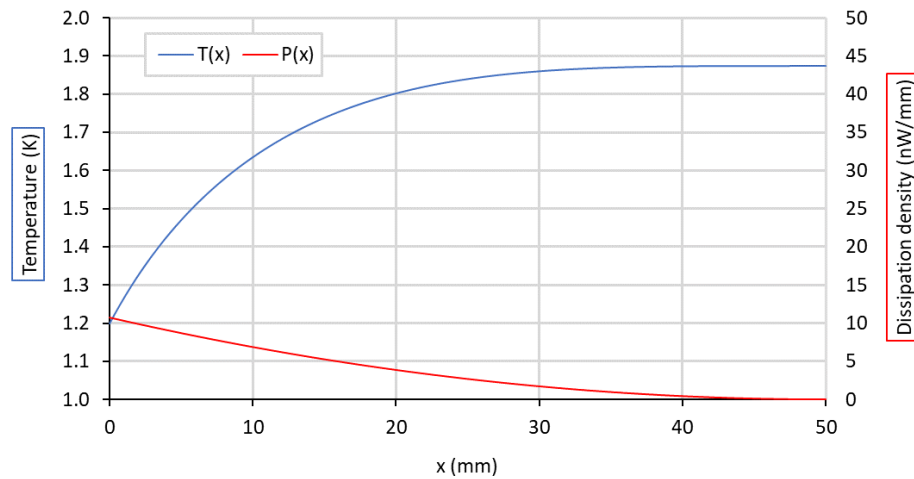


Figure C – Power law dissipation profile ($n=2$) with no heat load at heat switch tip ($P_{total} = 10 \mu\text{W}$)

Next, imposing the same total dissipation P_{total} and tip temperature T_2 , we study the impact of a redistribution of some of the dissipation to the tip on the deduce dissipation profile. When imposing a profile of $9 \mu\text{W}$ and a $1 \mu\text{W}$ heat load at the tip, the profile is of order 2.3. When imposing a profile of $8 \mu\text{W}$ and a $2 \mu\text{W}$ heat load at the tip, the profile order rises sharply to 7.4. A final case is tested, for a profile of $7.5 \mu\text{W}$ and a $2.5 \mu\text{W}$ heat load at the tip. The order of the profile then rises to 19.2, shown in Figure D.

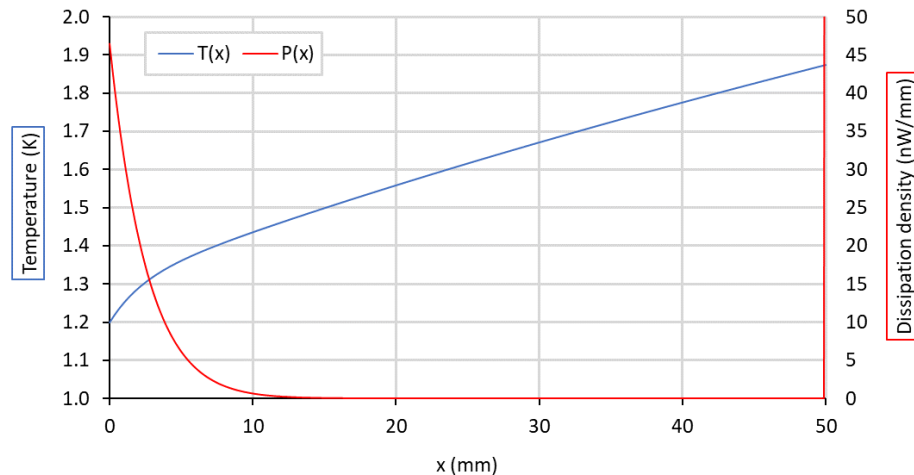


Figure D – Power law dissipation profile ($n=19.2$) with $2.5 \mu\text{W}$ heat load at heat switch tip ($P_{total} = 10 \mu\text{W}$)

The deduced n values for these 3 cases are summarised Table B.

Table B – Evolution of dissipation profile for increasing heat load at heat switch tip

	$P_{profile}$	P_{tip}	n (such that $T_2 = 1.875$ K)	x_c
0% at tip	10 μ W	0 μ W	2.0	12.5 mm
10% at tip	9 μ W	1 μ W	2.3	11.5 mm
20% at tip	8 μ W	2 μ W	7.4	5.5 mm
25% at tip	7.5 μ W	2.5 μ W	19.2	2.5 mm

For a dissipation profile described by a power law of the form $P(x) = A(L - x)^n$, we can deduce the weighted average dissipation location x_c from equation (4.10) in section 4.2.2.3. Shown in the last column of Table B, we note that when n is higher the dissipation location is closer to the base of the heat switch.

This confirms that for a given set of P_{total} and T_2 values, for an important heat load observed at the tip (as during experiments) microvibration dissipation along the heat switch is mostly concentrated at its base. This agrees with the assumption that dissipation is greatest where mechanical bending stress is too.

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Impact of microvibrations on low-temperature instruments dedicated to the study of the universe

For disruptive science missions – such as to study the hot and energetic universe or the cosmic microwave background – where detectors operate at very low temperatures (down to 50 mK), the thermal susceptibility to microvibrations (such as from reaction wheels, thrusters or cryocoolers) may become critical.

This thesis aimed to gain an understanding of this thermal susceptibility, for cryogenic devices and systems crucial for such missions. We built a highly sensitive test bench dedicated to studying dissipation due to microvibration disturbance. Using bespoke actuation and sensing methods, we were able to induce sub-millig levels of mechanical excitation and observe sub-microwatt levels of dissipation. We quantified dissipation in a gas-gap heat switch (a cryogenic device ubiquitous in sub-Kelvin space applications) and studied the locations and nature of the phenomena causing it.

Our work has largely been experimentally focused. It has also been backed by a simple thermomechanical dissipation model to interpret our measurements. For mechanical excitation around $1 \text{ mg}_{\text{rms}}$, we observed microvibration dissipation as high as $1 \text{ }\mu\text{W}$, at 1.2 K and 4.2 K. Large Q factors (>1000) of our heat switch (and variations thereof) were measured, allowing us to observe microvibration dissipation of flexible damped elements attached to our test object (e.g. instrumentation wiring) due to their forced oscillation.

Our work applies to cryogenic systems for low-temperature science missions, such as the future Athena X-ray space observatory, but also beyond space for quantum computers that also need very low and stable temperatures.

Impact des microvibrations sur les instruments très basses températures dédiés à l'étude de l'univers

Pour des missions scientifiques novatrices – telles que l'étude de l'univers chaud et énergétique ou du fond diffus cosmologique – dont les détecteurs fonctionnent à très basses températures (jusqu'à 50 mK), la sensibilité thermique aux microvibrations (par exemple de roues à réaction ou de cryoréfrigérateurs) est critique.

Cette thèse avait pour but de mieux maîtriser la sensibilité thermique des systèmes cryogéniques essentiels à ces missions. Nous avons conçu un banc d'essai sensible, dédié à l'étude de la dissipation due aux microvibrations. En utilisant des méthodes d'excitation et de détection spécifiquement développées pour nos travaux, nous avons pu induire des niveaux d'excitation mécanique inférieur au millième de g et observer des niveaux de dissipation inférieurs au microwatt. Nous avons quantifié la dissipation dans un interrupteur thermique à gaz d'échange (un dispositif cryogénique omniprésent dans les applications spatiales sub-Kelvin) et étudié la localisation et la nature des phénomènes qui la provoquent.

Notre travail a été principalement expérimental. Nous nous sommes aussi appuyé sur un modèle simple de dissipation thermomécanique pour interpréter nos mesures. Pour une excitation mécanique autour de $1 \text{ mg}_{\text{rms}}$, une dissipation due aux microvibrations atteignant $1 \text{ }\mu\text{W}$ a été observée, à 1,2 K et à 4,2 K. Des Q facteurs élevés (>1000) de notre interrupteur thermique (ainsi que des variations de celui-ci) ont été mesurés, ce qui nous a permis d'observer la dissipation d'éléments flexibles amortis accrochés à notre objet de test (par exemple, des fils de l'instrumentation) dû à leurs oscillations forcées.

Notre travail s'applique aux systèmes cryogéniques pour les missions scientifiques à basse température, tels que Athena, un futur observatoire spatial à rayons X, mais également au-delà du domaine spatial, par exemple pour les technologies pour les ordinateurs quantiques, qui nécessite des températures très basses et stables également.